PATENT

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Appl. No.	: 10/761,865		Confirmation No.:3149
Applicant	: Koneda et al.		
Filed	: January 21, 2004		
T.C./A.U.	: 3748		
Examiner	: Chang, Ching		
Docket No.	: 81044248		
Customer No.	: 33066		

DECLARATION UNDER 37 CFR 1.131

Mail Stop Amendment
T.C./A.U. 3748
Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

We, James Ervin, Thomas Megli and Philip Koneda, state that:

1. We are joint inventors of the above-identified patent application;

2. We made an invention claimed in the above-identified patent application in the United States prior to February 4, 2003.

3. In support thereof, we submit the following evidence:

A. A Document (see attached) created by the above-identified James Ervin entitled "DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER" by James D. Ervin, Thomas Megli and Philip Koneda, the date on said Document having been removed but such date being prior to February 4, 2003, said Document being attached as attached EXHIBIT A;

B. Said Document shows on page 3 in Figure 3 a hydraulic lever similar to FIG. 3 of the above-identified patent application;

C. On or about January 01, 2003, it was decide to initially fabricate the valve shown in Figure 3 of Exhibit A with the two pistons forming portions of the walls of

the chamber having the equal surface areas (i.e., a 1:1 ratio device). More particularly, we chose to make a 1:1 ratio device as a matter of convenience because it allowed us to utilize our pre-existing hardware and supporting test fixtures. Although this prototype was realized at 1:1, ALL of our modeling and analysis documentation was focused on characterizing behavior on ratios greater than 1:1. For our purposes, the real value of hardware is to validate the relationships being modeled and the 1:1 hardware was convenient for that purpose (confirm leakage / tolerance / viscous relationships, etc.). We then rely on the models to allow us to predict behavior at the ratio condition that is most appropriate for a given application, package space, etc.

D. A Document dated February 25, 2003 attached as EXHIBIT B, which are slides presented at a design review of the hydraulic lever described in the document in Exhibit A; designed at ratio 1:1. This hardware was fabricated between February 25th and July of 2003,

E . Exhibit C is a print out of a screen shot of a computer directory of the above-identified James Ervin showing that on March 17, 2003 a document was created entitled "patent review for hydraulic level and vibration cancellation.ppt";

F. Exhibit D is a copy of the document referred to in paragraph C above and entitled "patent review for hydraulic level and vibration cancellation.ppt";

G. Exhibit E is a copy of a shot of the above-identified Thomas Megli's computer screen showing analysis of the hydraulic lever shown on the second pages of the document in Exhibits D such shot such seen being shown dated in the upper right hand corner April 25, 2003.

H. Between April 25, 2003 and July 24. 2003 we focused on engineering assessment of various prospective suppliers. This work was essential as Ford relies on supplier partnerships to commercialize technology such as the hydraulic lever. In this search, the supplier is engaged at an exploratory level to assess whether the proper complement of skill, experience, and manufacturing capability are complementary with our long term need. The nature of these assessments can be gleaned though the file titles from this period from Exhibit "O": e.g. Actuator collision3.xls – evaluating the role of impacts between mechanical components, 6 bar open from M1.dat – evaluating the challenge of

actuating under gas forces, LSP JCAE comparison slide.ppt – internal high level system supplier comparison, supplier selection technical tradeoffs.ppt – internal technical system supplier comparison, TRW armature failure analysis translation.doc – internal component supplier evaluation, LSP power consumption estimate 072403 5.xls – projecting power consumption for comparative selection.

I. Exhibit F shows on the first page thereof a computer screen shot indicating that on July 24, 2003 computer aided design (CAD) was refined for the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.

J. Exhibit G is a copy of a page from a notebook of the above-identified Thomas Megli dated July 28, 2003 showing calculations related to the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.

K. Exhibit H is a copy of a page from a notebook of the above-identified Thomas Megli dated July 29, 2003 showing oil feed flow rate data measured for the prototype shown in exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.

L. Exhibit I is a copy of a page from a notebook of the above-identified Thomas Megli dated July 30, 2003 showing calculations related to the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.

M. During the period between February 25, 2003 and November 11, 2003, activity was towards building, testing, and refining a prototype of the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas. Typically the design process begins with a concept sketch that is reviewed with the CAD designer. The designer then converts the concept sketch into a realizable CAD cross-section that delivers the functions desired. This process of creation and review can take several weeks, and must consider how best to the implement the critical features that will be demonstrated with a hardware confirmation prototype. For the hydraulic lever, the critical feature to be

demonstrated was the ability to transmit force, thru the hydraulic path and return the armature to the fully released position. This was best implemented by converting existing magnetic circuit hardware from earlier prototype hardware. The design creation process, converting from sketch to realizable hardware typically takes about 1 month. After that the design is further refined to include secondary features, for the hydraulic lever this included tailoring the ports to ensure the oil transfer between the two chambers was stable. Once the design model was complete, we sent the model to a local machine shop who then built all of the details, from the model. This process took about 1 month. Once the hardware was received, we had to wait for an opportunity to test in our laboratory. This could have taken 1-2 months. Remember that our "internal concept development" was a secondary activity to our primary responsibility of identifying a suitable first tier supplier for this project. During assembly and debugging in July of 2003, we discovered that since the parts were made directly from the model, critical component dimensions, like the clearance between the hydraulic pistons and the housing was excessive and internal leakage was too high for adequate function. The piston dimensions were revised a new pistons fabricated adding another 1-2 weeks. Again, some delay to find time to test, and finally the testing that verified the feasibility of this concept. All in all, this additional time could have taken 2 to 4 months.

N. Exhibit J is a copy of a Management Performance Review reporting work on the hydraulic lever concept, including general modeling for ratios > 1:1 and testing of the 1:1 ratio device identified above. This was presented to our management on November 11, 2003 by the above-identified Thomas Megli with portions being removed.

O. Exhibit K is a copy of a Management Performance Review reporting work on the hydraulic lever concept, including general modeling and reporting for ratios > 1:1. This was presented to our management on November 11, 2003 by the above-identified James Ervin with portions being removed.

P. Exhibit L is a copy of a screen print of a listing of files having data taken on the above-identified 1:1 ratio device by the above-identified Thomas Megli dated July 2 and July 3 2003 .

Q. Exhibit M is a screen shot of the test data taken on the above-identified 1:1 ratio device taken on July 2 and July 3, 2003. Note that "Date Created" is later

in time because the files were copied from one computer to another. "Date Modified" dates show the last time modifications were made to the file.

S. Exhibit N are screen print outs showing creation of various files generated directly support the engineering development of the hydraulic actuator, various systems integration work (such as electrical system analysis, wiring analysis, magnetic material property research), and the EVA (electromagnetic valve actuation) supplier selection process (which was a critical step for commercializing EVA) described in Figure 3 of Exhibit A. Such files compiled for James Ervin's work as example activity, and being dated between the period of time between February 6, 2003 and January 24, 2004.

4. All statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true.

5. I understand and have been advised that willful false statements and the like made herein are punishable by fine or imprisonment, or both (18 U.S.C. 1001) and may jeopardize the validity of the application or any patent issuing thereon.

Date: 7/19/06 James Ervin

Date: 7/19/06 Thomas Megli

Date: 7/19/06 Philip Koneda

EXHIBIT A
(6 sheets)

DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER

James D. Ervin, Thomas Megli, Philip Koneda

Introduction

One common approach to controlled valve actuation is to use two electromagnets to toggle an armature connected to a valve between an open or closed position where it is held, while a pair of springs is used to force the valve to move (oscillate) to the other state (Fig. 1). A common shortfall to such a design is that the magnets must generate force over a travel length equal to the valve stroke. At points of travel where the air gap is large, more current is required to achieve a given force resulting in a corresponding increase in power consumption. Conversely, the peak force that can be generated for a given current is reduced as the air gap increases, which effectively reduces the authority to control the valve motion.

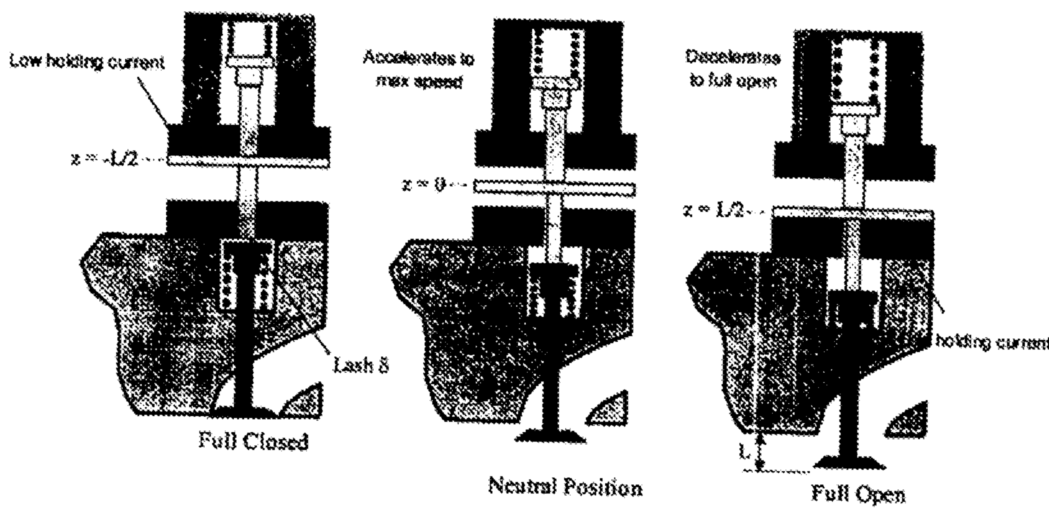


Figure 1: Conventional linear direct acting spring-mass oscillator

In contrast to the direct acting mass-oscillator approach, a new actuation architecture is proposed here which utilizes a hydraulic lever amplify the motion of a magnetic

armature to achieve a desired valve displacement, thus reducing the effective air gap. This disclosure describes the potential claims regarding this new architecture:

- Use of a hydraulic lever to minimize variation in air gap.
- Incorporation of passive hydraulic lash adjustment in the hydraulic lever mechanism.
- Incorporation of passive hydraulic damping in the hydraulic lever mechanism to potentially enable open loop control.
- Incorporation of a bypass valve to tune the damper response and allow for initialization of the system at startup.
- Ability to flexibly package the system to meet package requirements.

Prior art

As a departure from the direct acting systems, there is at least one example of a lever actuated system: LSP Innovative Automotive Systems of Munich, Germany has developed an actuator that uses a mechanical lever to amplify the travel of the armature and reduce the effective air gap. In contrast to the hydraulic system proposed here, the mechanical lever does not address issues of passive lash management or passive damping and must be designed within specific packaging rules. As a result of the reduced gap, the LSP lever system does improve the control authority through the stroke and improves the power consumption relative to conventional linear oscillators.

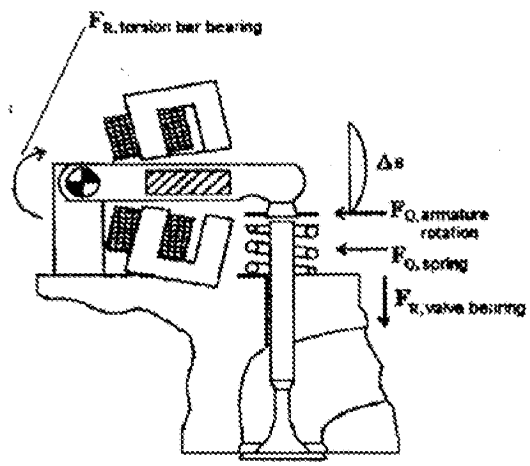


Figure 2: LSP lever acting spring-mass oscillator

Overview of the hydraulic amplifier lever oscillator

As the subject of this disclosure, a hydraulic lever is proposed as an alternative means of amplifying armature displacement. This has the effect of reducing the armature travel that is required to achieve a desired valve displacement and, in turn, reduces the effective air gap. In one proposed arrangement (Fig. 3), a first hydraulic piston is attached to an armature and biased with a first spring to be normally held in a downward position while a second piston is attached to a valve and biased with a second spring in a normally upward position.

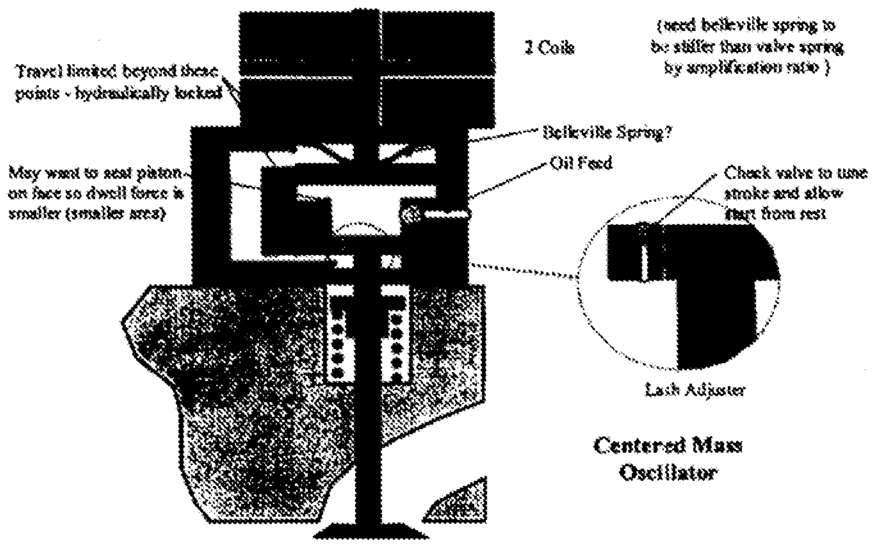


Figure 3: Hydraulic lever acting spring-mass oscillator

During a startup sequence, an upper coil is used to pull the armature upward. This creates higher pressure in the outer cavity than the central cavity to open a check valve mounted on the second piston, allow fluid transfer from the outer cavity to the central cavity, and compress the first spring.

Following this initialization process, the upper coil is de-energized, causing the first spring to move the armature and first piston downward. This increases the pressure in the central cavity between the piston faces, decreases the pressure in the outer cavity, and closes the check valve. The pressure difference across the second piston causes it to move downward and compress the second spring. At some time during this process, the lower coil is energized to continue compressing the second spring until the second piston strokes beyond the point where it can communicate oil with the outer cavity. At this

time, the second piston becomes hydraulically locked, travel stops, and the valve is held in the open position. Alternately, the travel limits could be regulated by designing the first piston to strike a shoulder in the central cavity or by driving the armature to land on the lower coil. are expected to be that is controlled by opposing coils.

Conversely, the lower coil can be de-energized and the upper coil can be energized to reverse the process and close the valve. It is expected that the pressure in the central cavity will be greater than that of the outer cavity during this event so that the check valve remains closed. This expectation requires that the force from the upper coil minus the force from the upper spring be less than the force of the second spring at all times and can be ensured by designing the preload of the second spring appropriately.

Lash Adjustment

The use of hydraulic lash adjusters for controlling tolerance stackup and thermal growth in valvetrains is well established. Each cycle following the seating of the valve, a controlled leak (orifice) from an oil lube reservoir is used to charge a hydraulic chamber to eliminate the clearance between the valve stem and the cam profile. In the case of EVA, this lash adjuster would be positioned between the valve stem and the actuation source. When acted upon by the cam, the oil becomes trapped in the cavity (can't leak out fast enough) and pushes on the valve with essentially no lash. The key to providing lash adjustment in this manner is designing the controlled leak to be slow enough to allow only a minimal change in length during the valve stroking process while being fast enough to account for the rate of length change due to thermal growth. It must also operate across a wide range of viscosity as the oil temperature.

Referencing the hydraulic lever implementation shown in Fig. 3, hydraulic lash control could be readily achieved by designing the appropriate clearance between the support body and the upper and lower pistons. In a typical sequence of operation, the lower piston would stroke upwards until the valve became seated. The upper piston would then continue stroking until its travel became limited by the seating of the armature or by a closing off of the channel that communicated oil to the underside of the lower piston (hydraulic locking). During this event, a differential pressure would develop across the upper piston, causing fluid to flow into the central cavity through the check valve and leak between the perimeter of the upper piston and the support body to resolve

the volume displaced. Having charged the cavity, the actuation source would be able to begin opening the valve with no lash.

During an opening event, downward motion of the upper piston would cause the lower piston to stroke downward, compressing the valve spring and creating a differential pressure across the lower piston. As a consequence of the differential pressure, fluid would flow out of the central cavity, resulting in a lower net stroke of the lower piston than for the upper piston. Such lost stroke is actually desired to account for valve growth due to thermal effects, where the loss of stroke (leakage) is ideally designed to be greater than the maximum thermal growth that can occur during a given cycle.

As a tradeoff during the valve closing event, the length lost during opening would result in the valve landing before the upper piston had finished stroking. With the natural coupling of position and velocity for the upper and lower pistons, it is advantageous to design the leakage to be as small as possible so that the travel of the two pistons is nearly the same. Thus if one is landed gracefully, both will, leading to simpler control.

Passive Damping

Taking advantage of the hydraulic architecture, it is also simple to incorporate passive damping into the actuator. As an example (Fig. 4), the travel of the pistons could be damped at the travel extremes by having the pistons close off a port, stopping the communication of oil to the rest of the circuit. At this point the system would be hydraulically locked though a conservation of volume. A check valve could be incorporated into the overtravel portion of the cavity to facilitate the release event. In a further refinement, the shape of the port could be tailored to provide a desired level of damping as a function of piston travel (ref. docket 201-1552 Variable Area Damper), as suggested in Fig. 4. Other implementations could include a ring extending from the piston and engaging mating cavity on the support (ram damper).

Passive, velocity-dependent damping offers significant advantages over active EVA control:

1. Reduces or eliminates the need for high speed, complex position and current feedback control of the EVA solenoids -- This complex control is presently required to achieve soft valve seating velocity for present actuator systems. The feedback control requires a high-speed (approximately 10-20kHz control loop

frequencies) computer, a position sensor for every engine valve/EMVA assembly. The control algorithms required for soft-landing are highly nonlinear, and require complex structures, such as adaptive or iterative learning control schemes to compensate for changes in actuator and valve characteristics over the life of the engine.

2. Improves system robustness and repeatability – The damper reacts to remove nearly all of the armature kinetic energy near the end of a valve transition; therefore, the damper achieves low contact velocities for a wide range of solenoid voltage control inputs. This robustness demonstrates that the damper will compensate for changes in operating characteristics due to manufacturing variability, engine wear, fluctuations in vehicle supply voltage, and changes in gas flow force disturbances.

As a drawback, such passive damping does dissipate energy and will increase power consumption relative to the undamped case. It should be noted however that active control methods to manage impact events, such as lash take-up and landing, increase power consumption as well. The magnitudes of these effects are still under investigation.

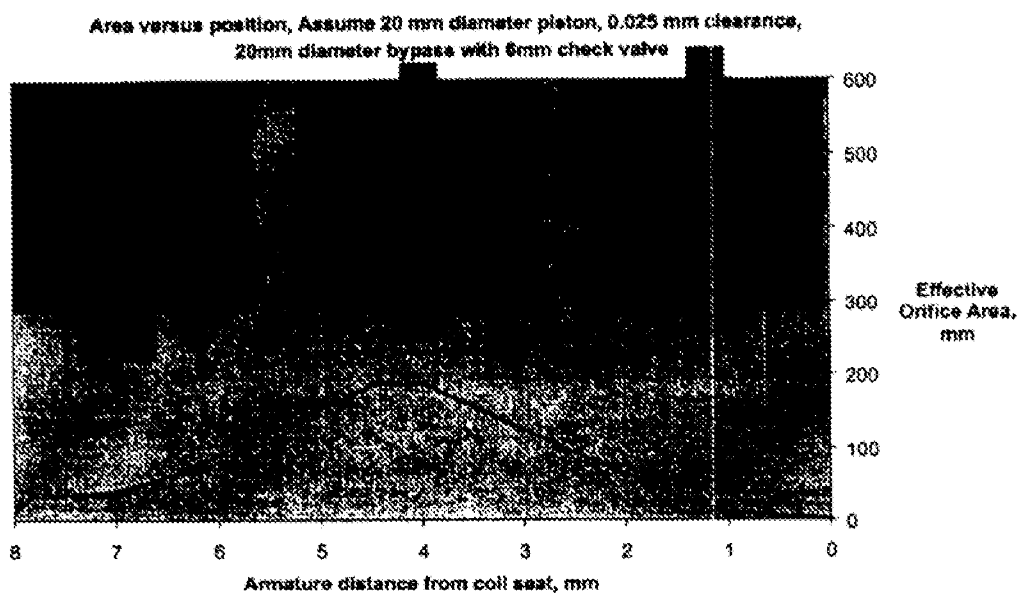


Figure 4: Example Implementation of Passive Damper

Vibration Cancellation

For mass-oscillating EVA systems, the typical valve acceleration and combined valve/armature mass is greater than for conventional cam designs leading to increased engine vibration. In an effort to minimize this effect, a design variant of the hydraulic lever is proposed where the upper port of the upper piston communicates with the upper port of the lower piston and the lower ports are similarly connected. This arrangement forces the motion of the two pistons in opposite directions, so that the inertial loads of each tend to cancel according resulting in a net force:

$$F_{net} = m_1 a_1 - m_2 a_2$$

where m_1 and m_2 are the masses of the upper and lower pistons and a_1 and a_2 are their respective accelerations.

Considering an example using armature and valve masses from conventional linear oscillator technology, a peak force of 630N would be produced by a 400g accel acting on the 0.090Kg upper mass (armature / stem / spring) and 0.073Kg lower mass (valve / keeper / retainers / spring). If the masses were driven in opposition by a hydraulic amplifier (1:1 amplification), the net force would instead be 66N, a tenfold reduction. If a larger amplification were considered, experience with the LSP lever system suggests that the armature mass would need to increase in keeping with the lever ratio while the associated acceleration would decrease.

Donut and hole opposed

Variable orifice damping

Find armature (upper) mass as a function of design parameters and valve (lower) mass:

$$\Delta t_{trans} = \pi\sqrt{\left(\frac{L \times SF \times TR}{2FPAM}\right)\left(\frac{M_v}{M_a} + \frac{1}{TR^2}\right)}$$

$$\frac{M_v}{M_a} = \frac{2 \times FPAM \times (\Delta t_{trans})^2}{L \times SF \times \pi^2 \times TR} - \frac{1}{TR^2} = \frac{2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2}{L \times SP \times \pi^2 \times TR^2}$$

$$M_a = M_v\left(L \times SF \times \pi^2 \times TR^2\right)\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2\right)^{-1}$$

Significant mass ratio relationships:

$$\frac{M_a}{M_v} = \frac{\left(L \times SF \times \pi^2 \times TR^2\right)}{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR)\right) - \left(L \times SF \times \pi^2\right)} = \frac{A}{B - C}$$

1) Lowest TR that can be applied for a given transition time (TR_{min}):

Armature mass goes to ∞ when TR satisfies the relationship:

$$B - C = 0 \Rightarrow TR \times sign(TR) = \frac{L \times SF \times \pi^2}{2 \times FPAM \times (\Delta t_{trans})^2}$$

Armature mass also goes to ∞ when TR goes to ∞

2) Find TR which produces the minimum armature mass (TR_{opt}):

Armature mass is minimum when:

$$\frac{\partial M_a}{\partial(TR)} = 0 = M_v\left(\frac{2\left(L \times SF \times \pi^2 \times TR\right)}{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2\right)} - \frac{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times sign(TR)\right)\left(L \times SF \times \pi^2 \times TR^2\right)}{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2\right)^2}\right)$$

$$\frac{\partial M_a}{\partial(TR)} = 0 = M_v\left(\frac{\left(2 \times L \times SF \times \pi^2 \times TR\right)\left(\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2\right) - \left(FPAM \times (\Delta t_{trans})^2 \times sign(TR) \times TR\right)\right)}{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times sign(TR) - L \times SF \times \pi^2\right)^2}\right)$$

$$\frac{\partial M_a}{\partial(TR)} = 0 \Rightarrow TR \times sign(TR) = \frac{L \times SF \times \pi^2}{FPAM \times (\Delta t_{trans})^2}$$

Notice that TR_{opt} is exactly twice the value of TR_{min}.

3) As TR is reduced from the TR for min armto the lowest applicable TR value, the effective mass is growing like

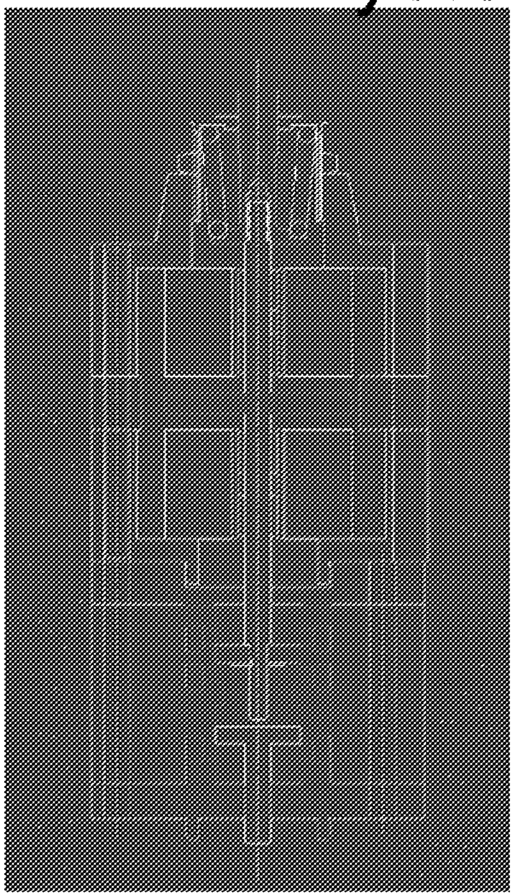
4) As TR approaches ∞ from the TR for minimum armature mass, the armature mass increases because more actuator force is required to deliver the same force to the valve.

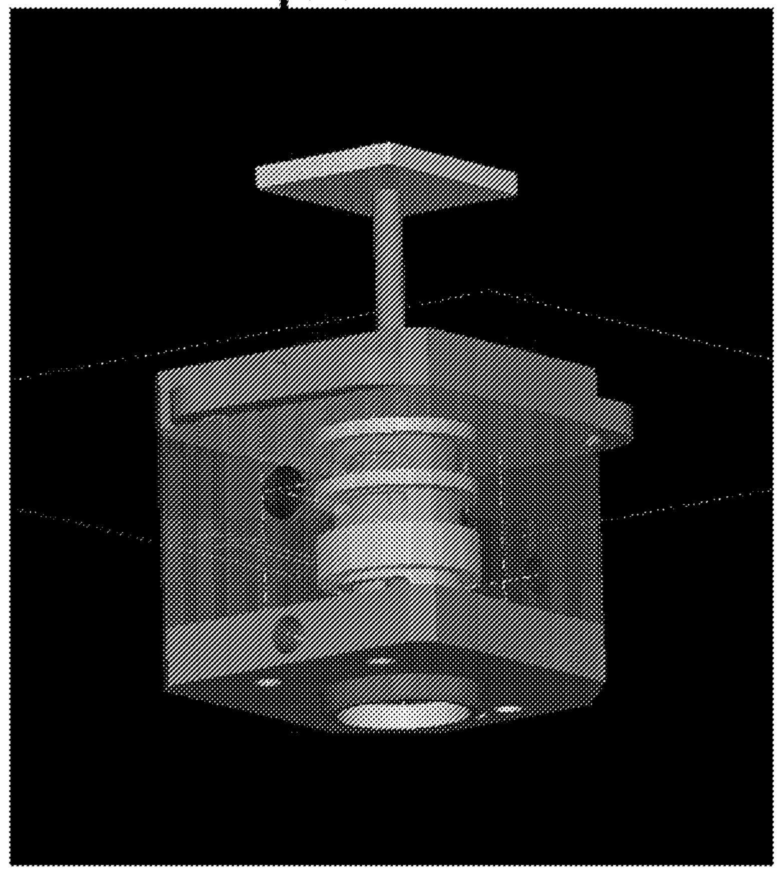
EXHIBIT B
(3 Sheets)

Actuator and Power Electronics Design and Engineering Review

Actuator Concept Development

Hydraulic lever update





ATDL flexible test rig for hydraulic lever with integrated damping

Research & Advanced Engineering

Ford Motor Company

February 25, 2003

Actuator Concept Development

Conclusions

- One mass damper results confirm earlier results documented contact velocities < 0.1 m/s
- Smaller diameter piston had better than expected impact velocity performance → may allow even smaller damper to further reduce mass and mid-travel losses
- Some unexplained temperature effects

Actuator Concept Development

Next Steps

- Complete testing and analysis of 1 and 2 mass dampers (ongoing with other ATDL testing)
- Build and test hydraulic lever prototype rig to prove out lash adjustment and damping functions (4-03)
- Initiate detailed package study for optimized system on AJ133 cylinder head(3-03)

EXHIBIT C
(1 Sheet)

hydraulic amplifier

hydraulic amplifier

	Name	Size	Type	Modified	Created
	203-0185 patent		File Folder	7/31/2003 1:...	7/17/2003 9:55 AM
	203-0185 base lever patent		File Folder	9/19/2003 11...	7/31/2003 1:17 PM
1	EVA Actuator with Hydraulic Displacement Amplifier.ppt	60 KB	Microsoft ...	4/10/2002 1:...	4/3/2002 3:29 PM
2	EVA Actuator with Hydraulic Displacement Amplifier2.ppt	225 KB	Microsoft ...	12/16/2002 5...	6/24/2002 12:40 PM
	Description of the Hydraulic Lever Invention.doc	20 KB	Microsoft ...	7/22/2002 11...	7/22/2002 11:57 AM
3	Description of the Hydraulic Lever Invention2.doc	1,271 KB	Microsoft ...	8/6/2002 2:1...	7/22/2002 12:28 PM
	Description of the Hydraulic Lever Invention3.doc	1,302 KB	Microsoft ...	10/3/2002 9:...	8/6/2002 2:16 PM
	Description of the Hydraulic Lever Invention4.doc	161 KB	Microsoft ...	11/7/2002 6:...	10/3/2002 10:58 AM
	Hydraulic Lever Invention amplification lash take up damping for patent.doc	1,258 KB	Microsoft ...	10/29/2002 1...	10/28/2002 10:59 AM
	Method to cancel vibrations in EVA actuators.doc	117 KB	Microsoft ...	12/16/2002 2...	10/29/2002 10:42 AM
	Hydraulic Lever Invention for controling two valves.doc	1,259 KB	Microsoft ...	10/30/2002 5...	10/30/2002 9:36 AM
	Method to optimize lever for min trans time.doc	134 KB	Microsoft ...	12/16/2002 3...	11/7/2002 6:03 PM
	88136_hyd lever lash take up damping disclosure.doc	1,267 KB	Microsoft ...	11/18/2002 8...	11/18/2002 8:32 AM
	Hydraulic lever EVA description for 88136 121002.doc	1,274 KB	Microsoft ...	12/16/2002 1...	12/10/2002 1:57 PM
	Driving multiple valves from one actuator 121602.doc	78 KB	Microsoft ...	1/2/2003 4:0...	12/16/2002 2:38 PM
4	patent review for hydraulic lever and vibration cancellation.ppt	1,400 KB	Microsoft ...	3/17/2003 12...	3/17/2003 12:03 PM
	97640MethodtocancelvibrationsinEVAactuators2.doc	154 KB	Microsoft ...	6/25/2003 4:...	6/25/2003 3:49 PM
	FGTI-080PUS-scanned figures 1-5.pdf	257 KB	Adobe Ac...	7/7/2003 11:...	7/7/2003 11:35 AM
	hydraulic amplifier patent application.doc	99 KB	Microsoft ...	7/7/2003 2:1...	7/7/2003 2:17 PM
	97640MethodtocancelvibrationsinEVAactuators3.doc	151 KB	Microsoft ...	7/9/2003 4:3...	7/9/2003 4:32 PM

How the "Hydraulic Lever Actuator" overcomes the shortcomings of Conventional "Direct-Acting Oscillator" Actuators:

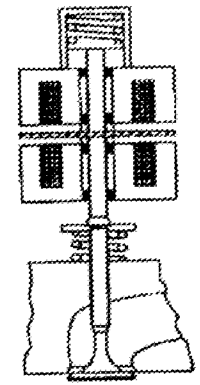
- Conventionally need electromagnets to act over large working gap
  -> inefficient
  -> large inductive change = hard to control
  ✓ lever reduces working gap

- Transition time is limited by fundamental laws of force / armature mass and stiffness
  -> engine performance is limited
  ✓ lever reduces effective mass – faster transition time

- Closed loop control of "soft landing" is difficult
  -> questionable robustness / reliability
  ✓ incorporate damping into hydraulic circuit to improve control of landings

- Mechanical lash requires use of energy consuming soft-release strategy
  -> inefficient
  ✓ incorporate hydraulic lash take-up

Direct acting spring-mass oscillator



Hydraulic lever acting spring-mass oscillator

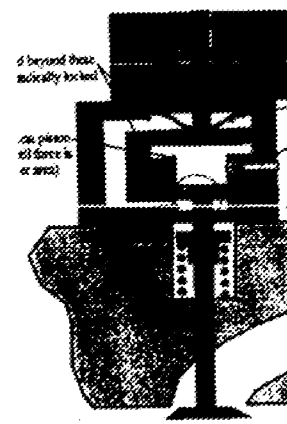


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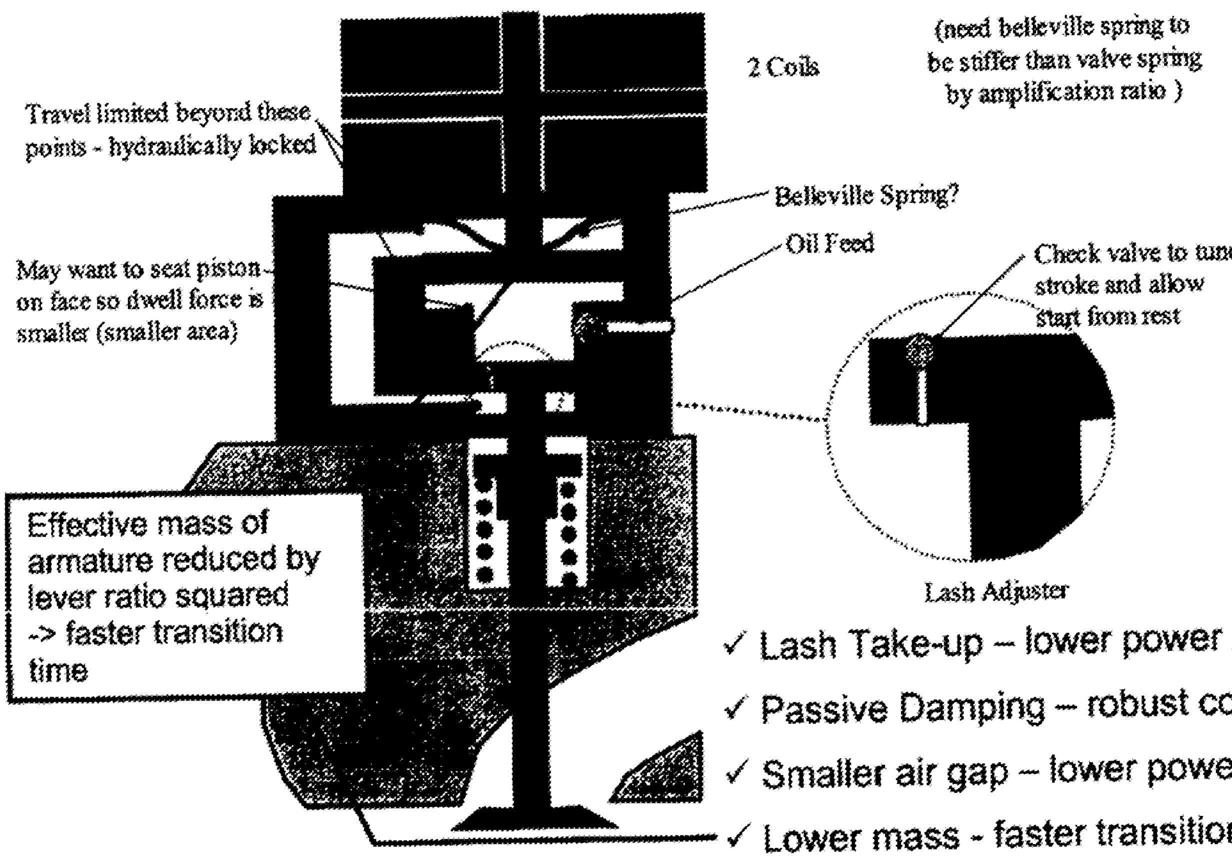
EXHIBIT D
(7 Sheets)

A Proposed Implementation of the Hydraulic Lever Oscillator

Hydraulic lever acting spring-mass oscillator



Effective mass of armature reduced by lever ratio squared -> faster transition time

- ✓ Lash Take-up – lower power / NVH
- ✓ Passive Damping – robust control / NVH
- ✓ Smaller air gap – lower power consumption
- ✓ Lower mass - faster transition times

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Effect of Lever on Magnetic Efficiency

Direct acting has lower consumption during hold since lever is a force divisor, but holding consumption is less than 10% of total consumption

Lever reduces working gap to reduce current required to generate desired force.

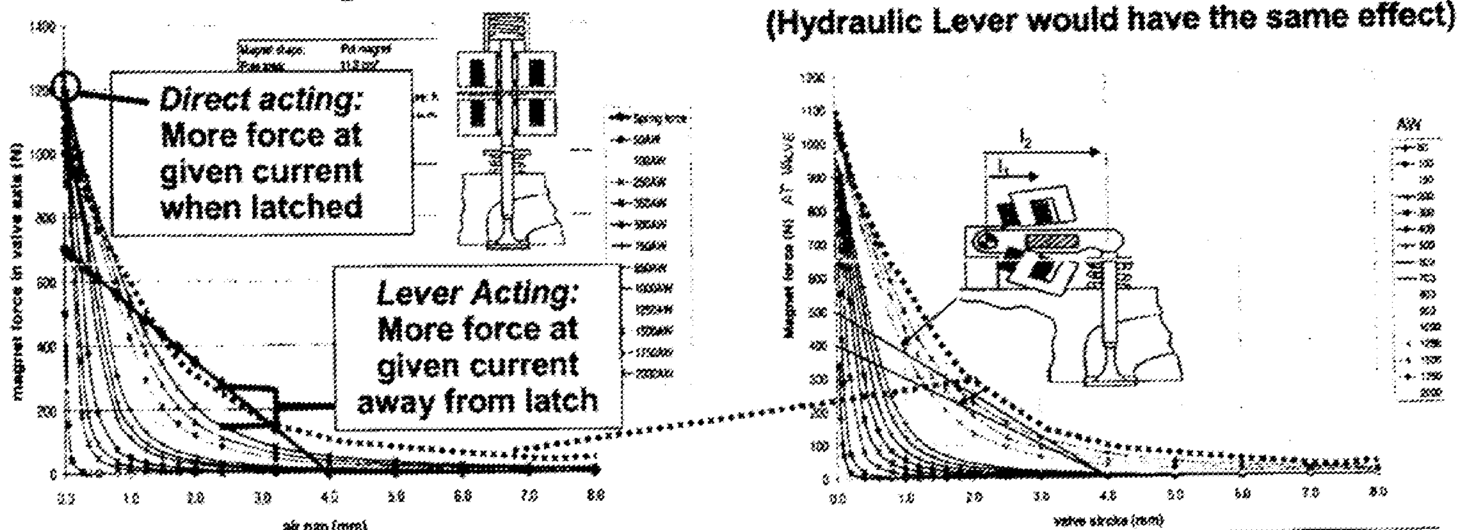
Current reduces by a factor of the lever ratio for modest to large gaps to reduce consumption during transitions.

Net power savings can be substantial – more than 50% for lever ratio > 2.

Direct Acting Oscillator

LSP Lever Actuator

(Hydraulic Lever would have the same effect)



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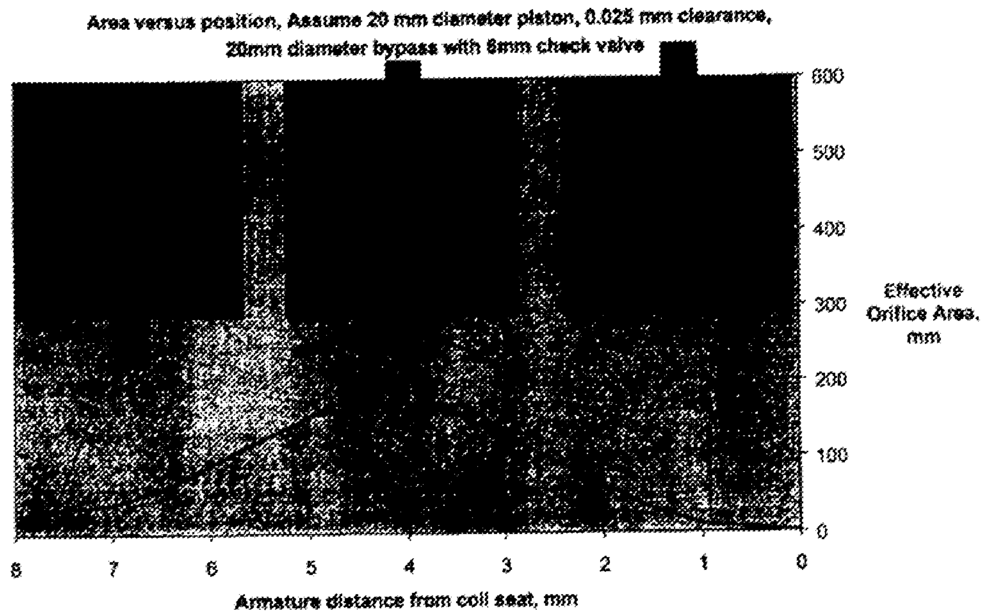
Passive Damping

Passive Damping Implementation Example:



Research
Ford Motor Company

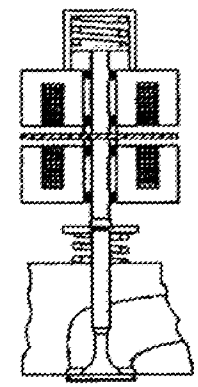
Vibration cancellation in EVA actuators through "opposed masses" in a Hydraulic Lever Actuator

- Conventional EVA actuators generate large shaking forces due to relatively high mass and fast transition time

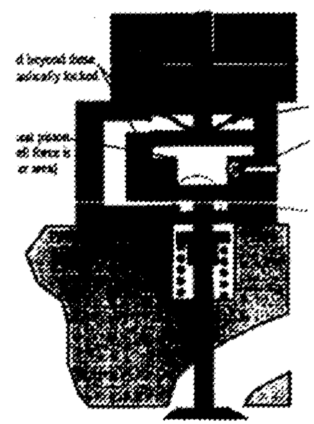
  -> much higher shaking forces than non-EVA valvetrain – especially at idle
  -> can't reduce transition time without functional loss
  -> can't reduce mass due to physical limitations

✓ Hydraulic lever can be configured to allow armature and valve to move in opposite directions to produce net force reduction / elimination

Direct acting spring-mass oscillator

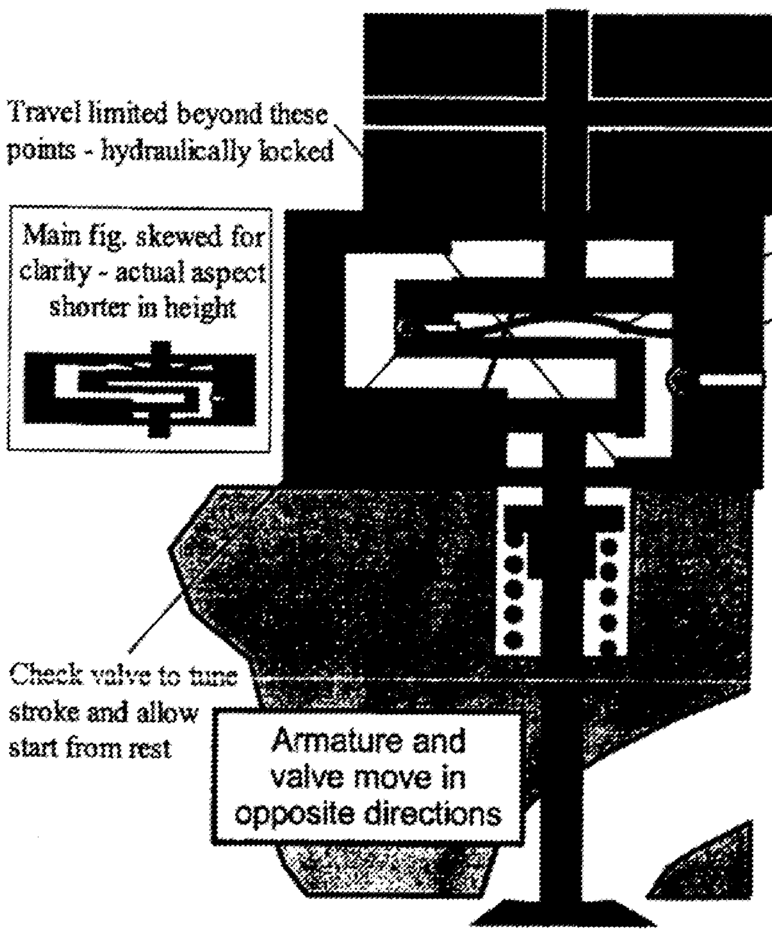


Hydraulic lever acting spring-mass oscillator



Research
Ford Motor Company

A Proposed Implementation of the Countermotion Hydraulic Lever Oscillator



- Masses move in opposite directions to cancel out vibrations internally

- Piston clearance provides leakage to resolve lash.

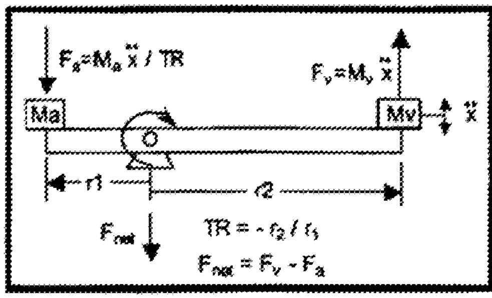
✓ Mass opposed
-> lower shaking forces
-> improved NVH

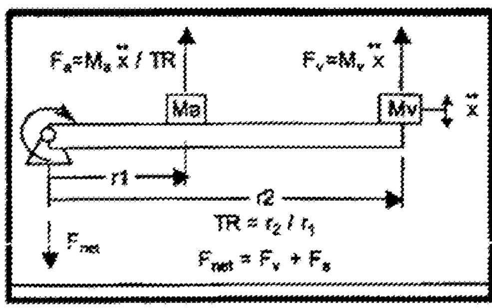
Research
Ford Motor Company.

Quantitative Vibration Force Improvement

<u>Example For 2.9ms Transition:</u>

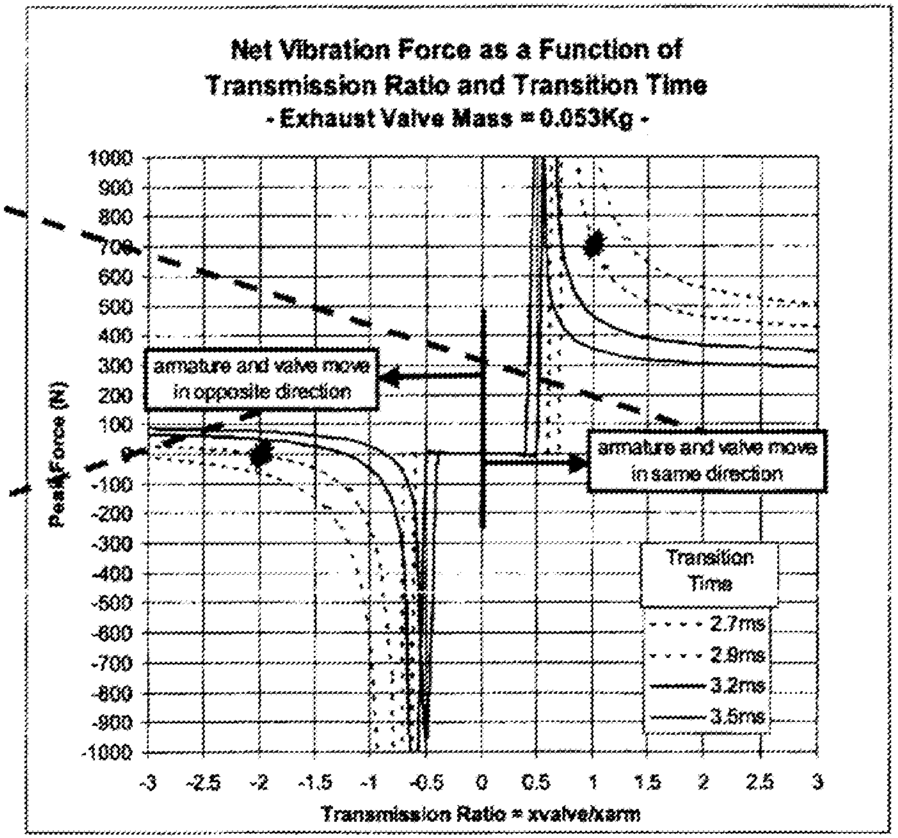
Reduce forcing from 700N (direct acting) to 0N (-2 lever ratio)!





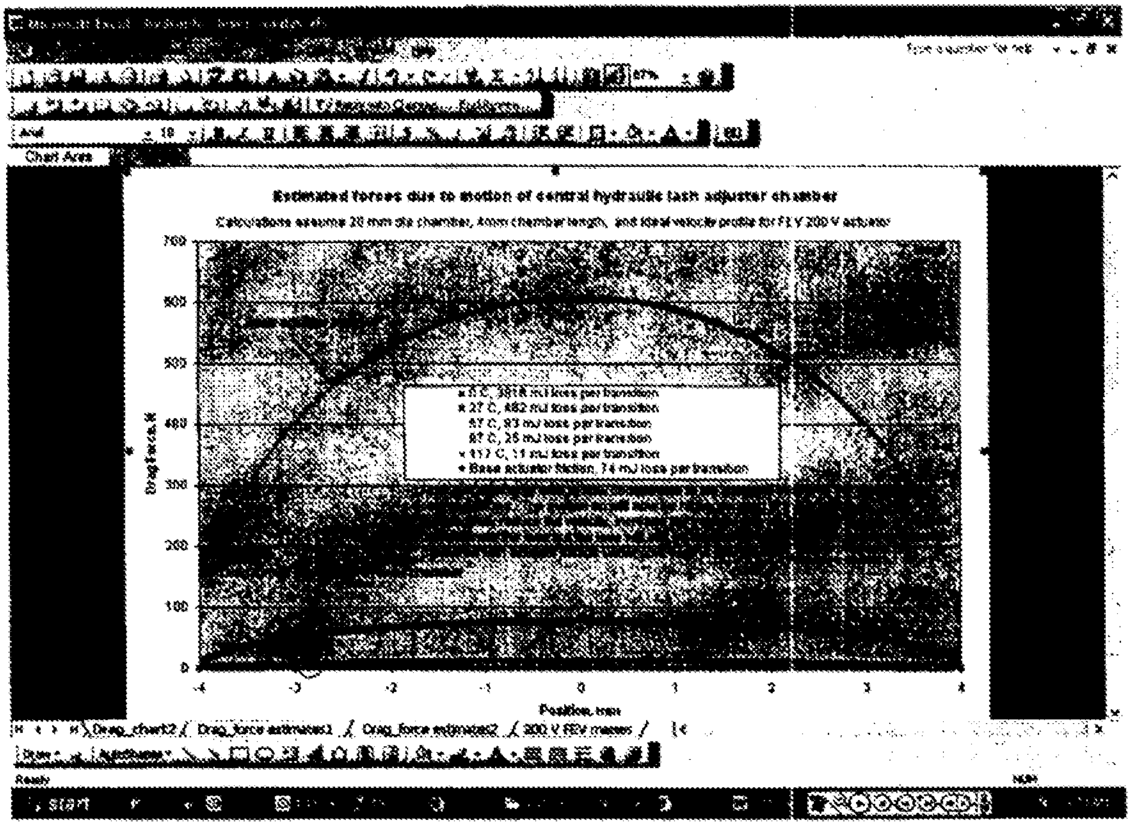
Nomenclature for First (upper) and Third (lower) Class levers

Example Net Vibration Force as a Function of Transmission Ratio

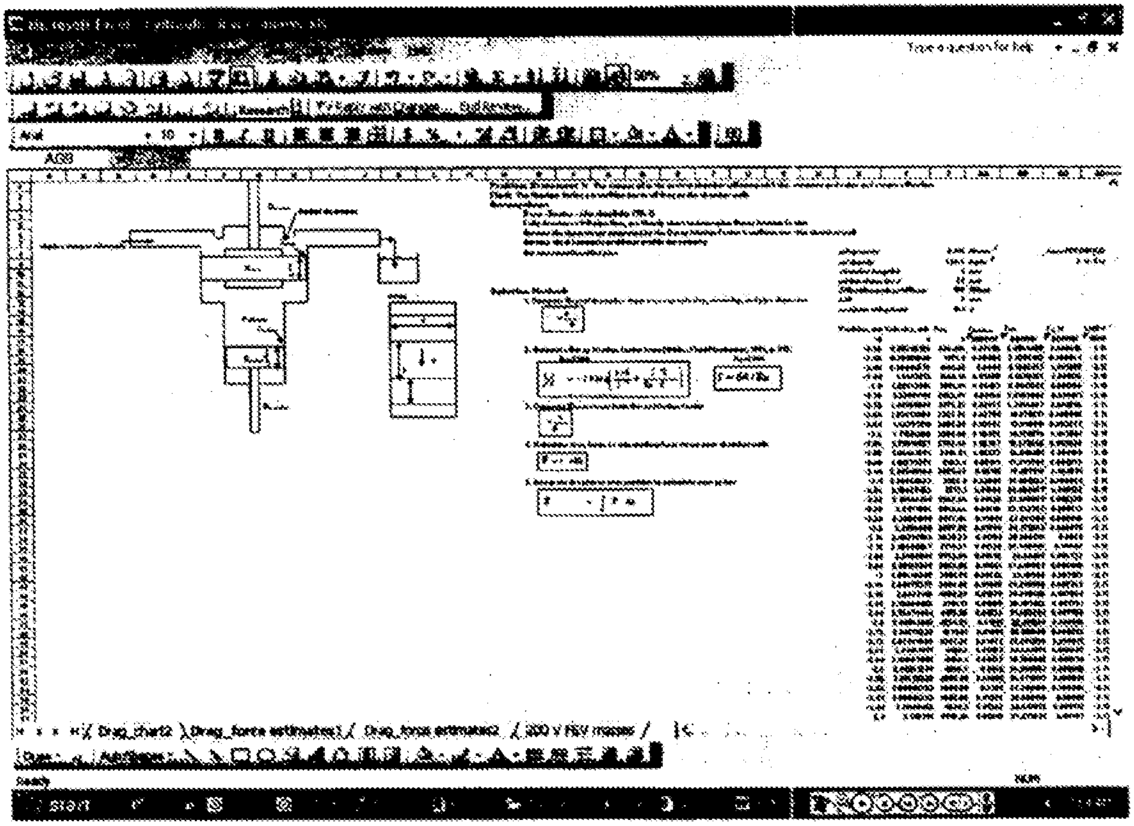


Research
Ford Motor Company

EXHIBIT E
(2 Sheets)



Design analysis worksheet

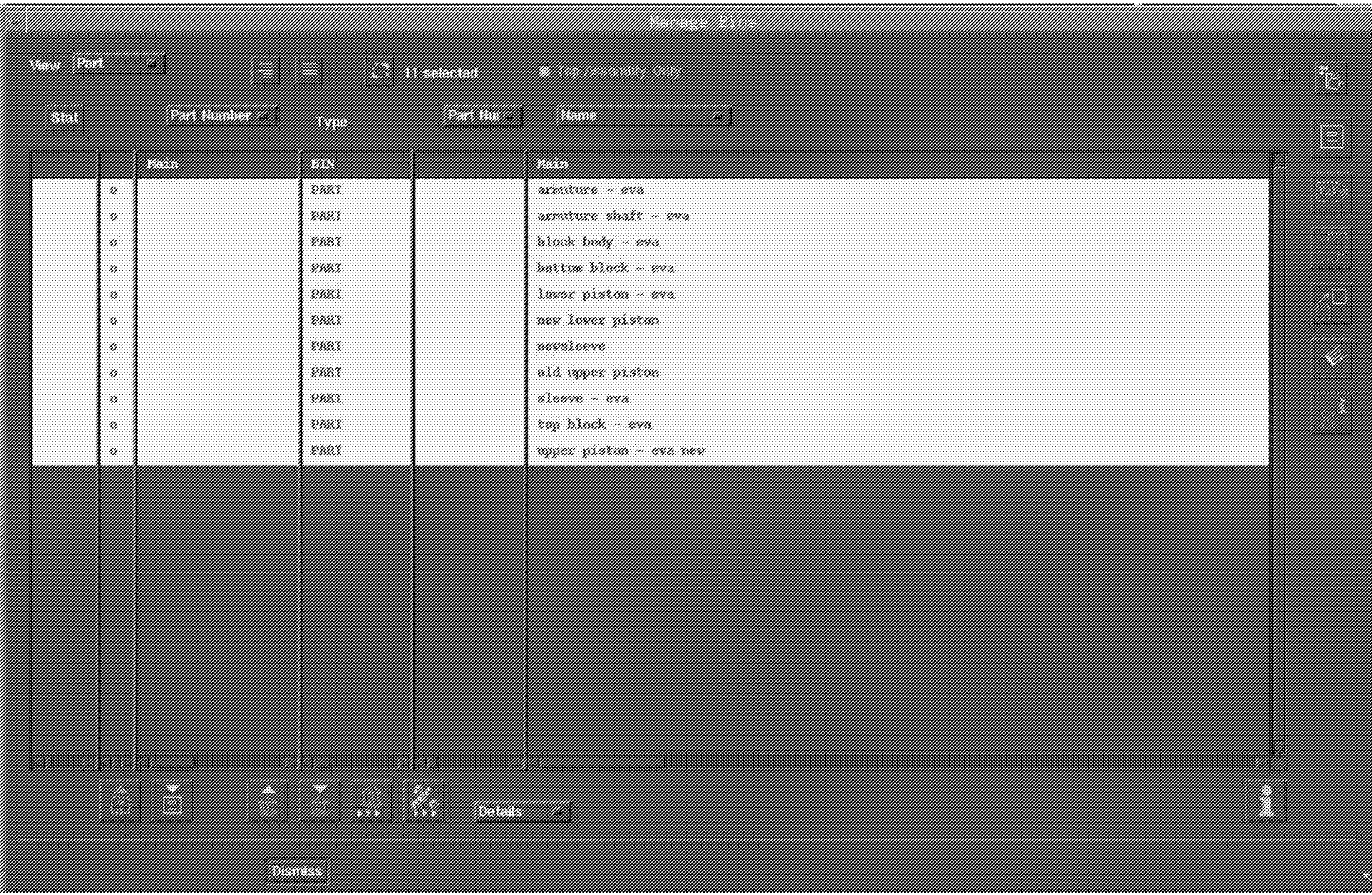


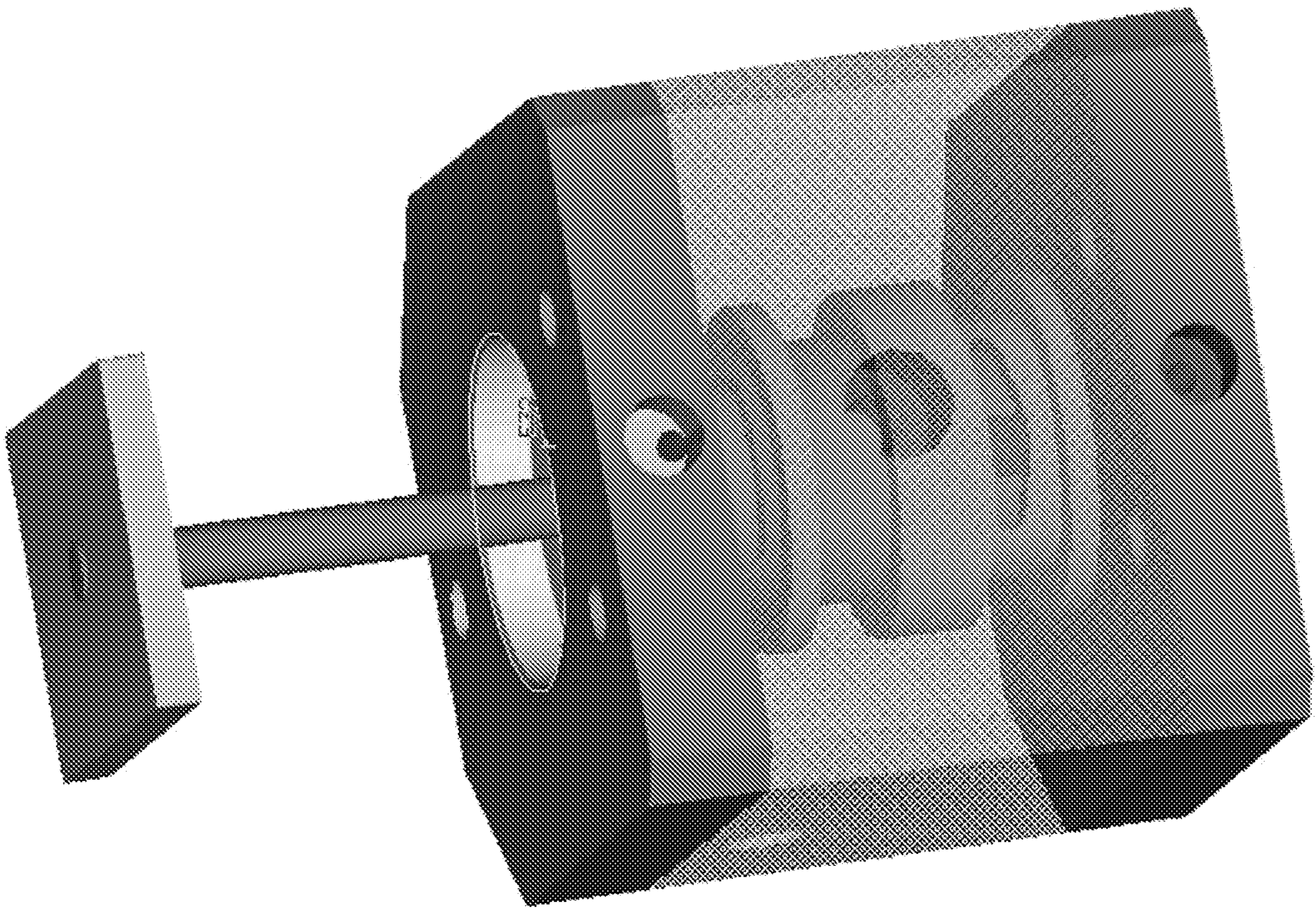
Design analysis worksheet

/ford/thishost/proj/ideas/sagdorny

EXHIBIT F
(3 Sheets)

oi5115.aux	Jun 21 14:11:08 2004	24576	-rw-rw----	sagdorny	sagdorny
ideas18153.und	Jun 18 09:39:22 2004	0	-rw-rw----	sagdorny	sagdorny
ideas9239.und	Jun 16 06:52:21 2004	0	-rw-rw----	sagdorny	sagdorny
CIValMsgsPart:CL8A53 (...)	Jun 15 13:26:43 2004	956	-rw-rw----	sagdorny	sagdorny
CIValMsgsPart:CL-8A5 (...)	Jun 15 13:12:33 2004	369	-rw-rw----	sagdorny	sagdorny
ideas7170.aux	Jun 14 07:38:05 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas7170.und	Jun 14 07:37:41 2004	0	-rw-rw----	sagdorny	sagdorny
ideas5090.aux	Jun 14 07:32:47 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas5090.und	Jun 14 06:55:34 2004	0	-rw-rw----	sagdorny	sagdorny
ideas15113.aux	Jun 11 14:19:10 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas15113.und	Jun 11 12:44:36 2004	0	-rw-rw----	sagdorny	sagdorny
oi15154.aux	Jun 11 12:04:33 2004	24576	-rw-rw----	sagdorny	sagdorny
ideas5086.aux	Jun 07 09:23:50 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas5086.und	Jun 07 09:20:59 2004	0	-rw-rw----	sagdorny	sagdorny
oi5128.aux	Jun 07 08:04:26 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas5240.aux	Jun 03 13:22:48 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas17936.aux	May 28 07:17:30 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas17936.und	May 28 07:04:34 2004	0	-rw-rw----	sagdorny	sagdorny
ideas11639.aux	May 27 14:27:54 2004	8192	-rw-rw----	sagdorny	sagdorny
userApp.cfg	May 27 11:27:44 2004	0	-rw-rw----	sagdorny	sagdorny
ideas11639.und	May 27 11:27:34 2004	0	-rw-rw----	sagdorny	sagdorny
oi11680.aux	May 26 14:04:57 2004	24576	-rw-rw----	sagdorny	sagdorny
ideas9591.aux	May 25 07:10:40 2004	8192	-rw-rw----	sagdorny	sagdorny
ideas9591.und	May 25 07:09:28 2004	0	-rw-rw----	sagdorny	sagdorny
sagdorny_eva0703.arc	Jul 24 08:00:00 2003	6923511	-rwxrwxrwx	sagdorny	sagdorny
vmaxpart.arc	Apr 01 09:27:00 2003	17484334	-rwxr-----	sagdorny	sagdorny


Manage EINs
View Part
11 selected
Top Assembly Only
Stat
Part Number
Type
Part Nur
Name
Main
EIN
Main
PART armature - eva
PART armature shaft - eva
PART block body - eva
PART bottom block - eva
PART lower piston - eva
PART new lower piston
PART newsleeve
PART old upper piston
PART sleeve - eva
PART top block - eva
PART upper piston - eva new
Details
Dismiss

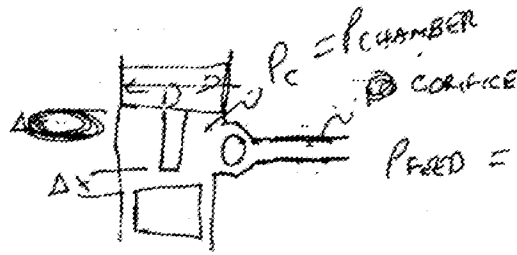


From T. Meglis Lab Notebook

7-28-03

7-9-08

Question: What is the characteristic fill time of air lever system when pulling?



Assume Bernoulli orifice flow

$$\frac{P_c}{\rho} + \frac{V_c^2}{2} + \cancel{g z_c} = \frac{P_f}{\rho} + \frac{V_f^2}{2} + \cancel{g z_f}$$

~~Assume~~ $\cancel{V_c \ll V_f}$

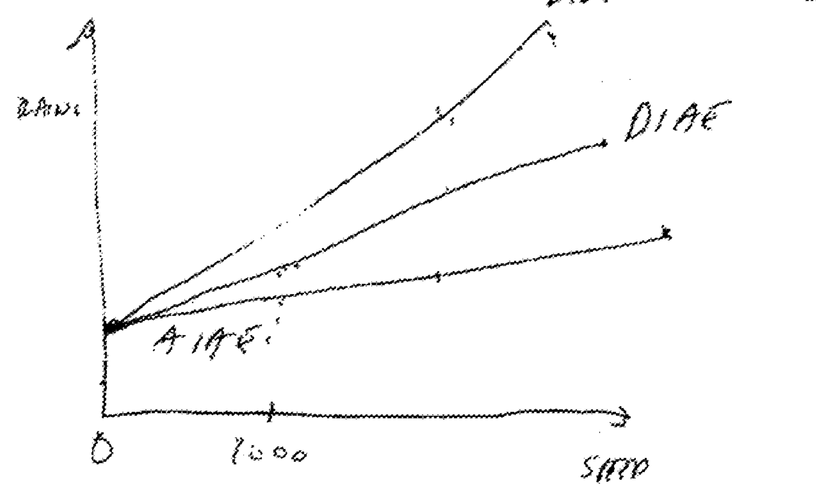
$$\Rightarrow \cancel{\frac{V_f^2}{2} = \frac{P_c - P_f}{\rho}}$$

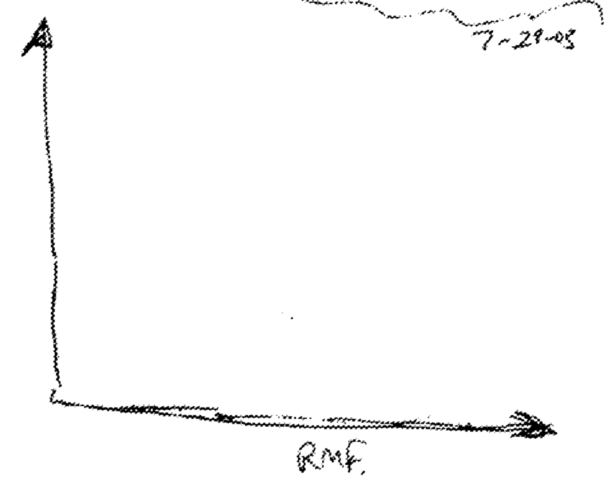
$$\Rightarrow \cancel{V_f = \sqrt{\frac{2(P_c - P_f)}{\rho}}}$$

$Q = V_c A_c = V_f A_f$ Continuity for incompressible flow

$$\Rightarrow \frac{P_c}{\rho} + \frac{Q^2}{2A_c^2} = \frac{P_f}{\rho} + \frac{Q^2}{2A_f^2}$$

EXHIBIT G
(2 Sheets)





71

From T. Meglis

Ques:

Time ?

Assume

Assume

⇒ /

⇒ X

G

⇒ $\frac{P_c}{e}$

EXHIBIT H
(2 Sheets)

7-29-03

CHECK OIL FLOW RATE THROUGH DAMPER CHECK VALVE UNDER VARIOUS PRESSURE

$\rho = 7.1$ lbs/gal

ΔP	$\dot{Q}$	
28 PSIG	0.001 GPM	MINIMUM } FULL PLUMBING THROUGH CHECK VALVE
30 PSIG	0.014 GPM	
80 PSIG	0.024 GPM	
28 PSIG	0.001 GPM	~~[illegible]~~ REMOVE HOSE FROM CHECK VALVE
50 PSIG	0.012 GPM	
80 PSIG	0.022	
20 PSIG	0.6 GPM	REMOVED HOSE BACK TO QUICK DISCONNECT
20 PSI	0.22 GPM	ADDED LINE BACK W/ NEEDLE VALVE
28 PSI	0.28 GPM	} REPLUMBED TO CHECK VALVE
20 PSI	0.19 GPM	
50 PSI	0.48 GPM	

553 /

$$\frac{\Delta V}{Q} = \frac{\pi D^2 \Delta x}{A_c \sqrt{\frac{2 \Delta P}{\rho}}}$$

$$= \frac{\pi D^2 \Delta x}{\frac{\pi d^2}{4} \sqrt{\frac{2 \Delta P}{\rho}}}$$

$$= 4\left(\frac{D}{d}\right)^2 \frac{\Delta x}{\sqrt{\frac{2\Delta P}{\rho}}}$$

$\approx \left(\frac{20}{2}\right)^2$ NEED TO CHECK

$\Delta x = 1\,mm$ ROUGH GUESS!
$\Delta P \approx 300000$
$\rho \approx 700\ \frac{kg}{m^3}$

$$\tau = \frac{4(100)\ .001\,m}{\sqrt{\frac{2(300000)}{700}}} \approx \sqrt{1000} \approx 30$$

$$\tau \approx \frac{400\,(.001)}{\cancel{[illegible]}20} \approx \frac{.4}{30}$$

75

7-30-03

DFSS CHARACTERIZE PHASE WORKSHOP

~~DECOUPLING~~ COUPLING ~~TECHNIQUES~~ ANALYSIS
EDITION
YAN WANG

S
PLOTE CAUSE AND EFFECT DIAGRAM
- CATEGORIZE CONTROL, NOISE, SIGNAL
- PRIORITIZING
INCLUDES SCORING OUT
CRITERIA
(1) IMPACT ON Y
(2) FEASIBILITY
↓
SCOPE
- P-DIAGRAM
ANALYZE FOR MODELING CAPABILITY
- EXPERIMENTAL SET-UP

$Y = f(x)$

EXHIBIT 1
(2 Sheets)

30-02

SIGNATURE

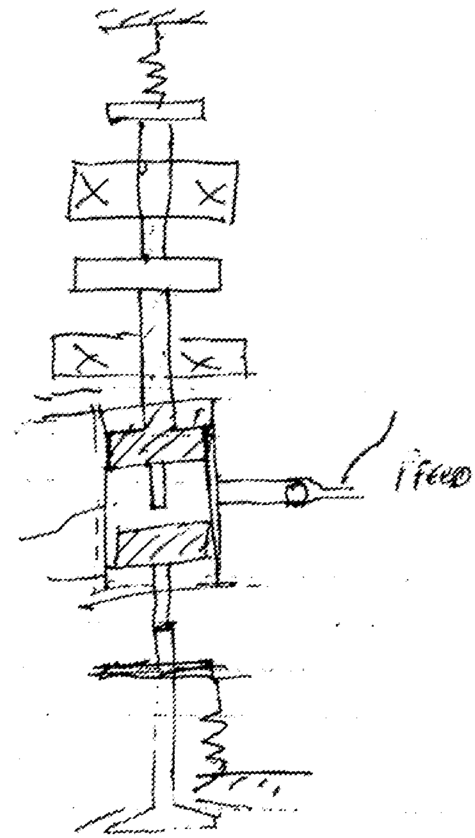


EXHIBIT J
(3 Sheets)

Ford Motor Company **Management Performance Review**

Name: Thomas W. Megli **Position:** Technical Specialist

National/Global ID: [redacted] **Performance Period:** 2003

[illegible] TO MOVE YOUR CURSOR BETWEEN SECTIONS, DO NOT USE THE TAB KEY

KEY BUSINESS OBJECTIVES

Competitive Assessment and IR Supplier Identification and Development:

A. Support competitive assessment of actuator package and transition time flexibility through the development and refinement of spring mass oscillator scaling models (Q4)

B. Establish fuel economy trade-offs of alternative actuator designs and valve operating modes using power consumption measurements and minimap analysis (Q1)

C. Complete the assessment of variable lever intermediate device (MFF concept) (Q2)

D. Support IR supplier development by identifying performance improvement opportunities.

Internal Concept Design and Development:

Lead design effort for hydraulic lever actuator concept

A. Develop dynamic models and use to establish design parameters (Q1)

B. Complete testing of armature damper prototypes to document performance and establish integrated damper design parameters (Q2)

C. Build and test hydraulic lever actuator prototype to demonstrate function, refine models, and quantify performance benefits (Q1 and Q2)

D. Design optimized actuator based on workhorse results (Q3 and Q4)

E. Provide engineering support and leadership to designers for concept development.

[redacted]

CONTRIBUTIONS TOWARD OBJECTIVES

Competitive Assessment:

A. Completed study to estimate relative FE benefits of lever and direct acting EVA systems and to investigate tradeoffs between peak actuator power, valve operation strategy, and drive cycle FE. Lever style actuator found to offer 2% M-H FE benefit over direct acting system. Flexible valve deactivation found to offer 2-4% FE benefit over dual valve operation.

Internal Concept Design and Development:

A. Completed Dymola system model to couple hydraulic lever submodels to EVA actuator submodel. The model was used to design the prototype piston diameters, clearances, and lengths for proper lash adjustment bleed down and damping characteristics.

B. Completed testing of one mass and two mass damper prototypes and documented contact velocity and transition time performance as a function of diameter and oil temperature. Made design modifications to reduce friction and improve design stack-up for further testing.

C. Completed build of hydraulic lever prototype hardware for FEV 200V system. Testing to begin 6-9-2002.

[redacted]

For further comments, go to the additional comments page attached

ASSESSMENT OF LEADERSHIP BEHAVIORS *: Based on feedback on Ford leadership behaviors*

Ford Motor Company **Management Performance Review**

Name: Thomas W. Megli | Position: Technical Specialist

National/Global ID: [redacted] | Performance Period: 2002

USE YOUR MOUSE TO MOVE YOUR CURSOR BETWEEN SECTIONS. DO NOT USE THE TAB KEY.

KEY BUSINESS OBJECTIVES

[redacted]

New actuator concept development (40%):
Develop alternative actuator concepts with emphasis on hydraulic damper design. (1) revise and calibrate model and evaluate redesign to deliver function with minimal losses. (2) deliver revised design specifications and concepts to hardware design and build team. (3) Support hardware testing and evaluation (Q4).

Other (20%):

A. Support Dymola camless wall-wetting model development (Q3).
B. NVH torsional analysis and induction noise support

CONTRIBUTIONS TOWARD OBJECTIVES

Competitive Assessment:

[redacted]

C. Developed analytical E-core force model and implemented to study damper control designs. Continuing model development to add features to support design of new actuator concepts. Completed engine cycle simulations to predict gas forces on exhaust valve during opening as a function of speed, load, and timing to support disturbance rejection analysis. Models have been correlated against test data.

New actuator concept development:

A. Developed concept and led the design of a 2-mass damper to achieve low impact velocities with minimal impact on transition time and lash compensation; patent application filed; hardware completed testing to begin in November.

(continued on additional comments sheet)

For further comments, go to the additional comments page attached

ASSESSMENT OF LEADERSHIP BEHAVIORS *(Based on feedback on Ford leadership behaviors)*

[redacted]

[redacted]

Additional Comments

KEY BUSINESS OBJECTIVES	CONTRIBUTIONS TOWARD OBJECTIVES
	New actuator concept development (continued). B. Developed hydraulic damper bypass control concept that schedules landing velocity as a function of engine speed and oil temperature. Validated with hardware and and submitted invention disclosure. [redacted] D. Developed concepts for integrating hydraulic damping function into hydraulic lever actuator concept. 2 invention disclosures submitted and hydraulic lever design initiated. [redacted] F. Developed actuator dynamics model to support hydraulic lever actuator design. Other: [redacted]

ASSESSMENT OF LEADERSHIP BEHAVIORS *(Based on feedback on Ford leadership behaviors)*

EXHIBIT K
(2 Sheets)

Ford Motor Company **Management Performance Review**

Name: James D. Ervin **Position:** LL6 - Technical Expert

National/Global ID: **Performance Period:** 2003

USE YOUR MOUSE TO MOVE YOUR CURSOR BETWEEN SECTIONS. DO NOT USE THE TAB KEY.

KEY BUSINESS OBJECTIVES	CONTRIBUTIONS TOWARD OBJECTIVES

For further comments, go to the additional comments page attached

ASSESSMENT OF LEADERSHIP BEHAVIORS *(Based on feedback on Ford leadership behaviors)*

Additional Comments

KEY BUSINESS OBJECTIVES	CONTRIBUTIONS TOWARD OBJECTIVES
> Contribute to development of internal actuation concept to improve function and quality while reducing cost (Q2). Support development of a production oriented system (Q4).	> Prepared documentation of hydraulic lever concept including disclosures and patent applications (Q2-3).
> Develop Ford IP and submit disclosures. (Q4)	> Prepared patent applications for hydraulic lever and vibration cancellation and disclosures for minimum hold power and valve mask with mini lift. VCR disclosure submitted. TCVT patent issued.

ASSESSMENT OF LEADERSHIP BEHAVIORS *(Based on feedback on Ford leadership behaviors)*

EXHIBIT L
(1 Sheet)

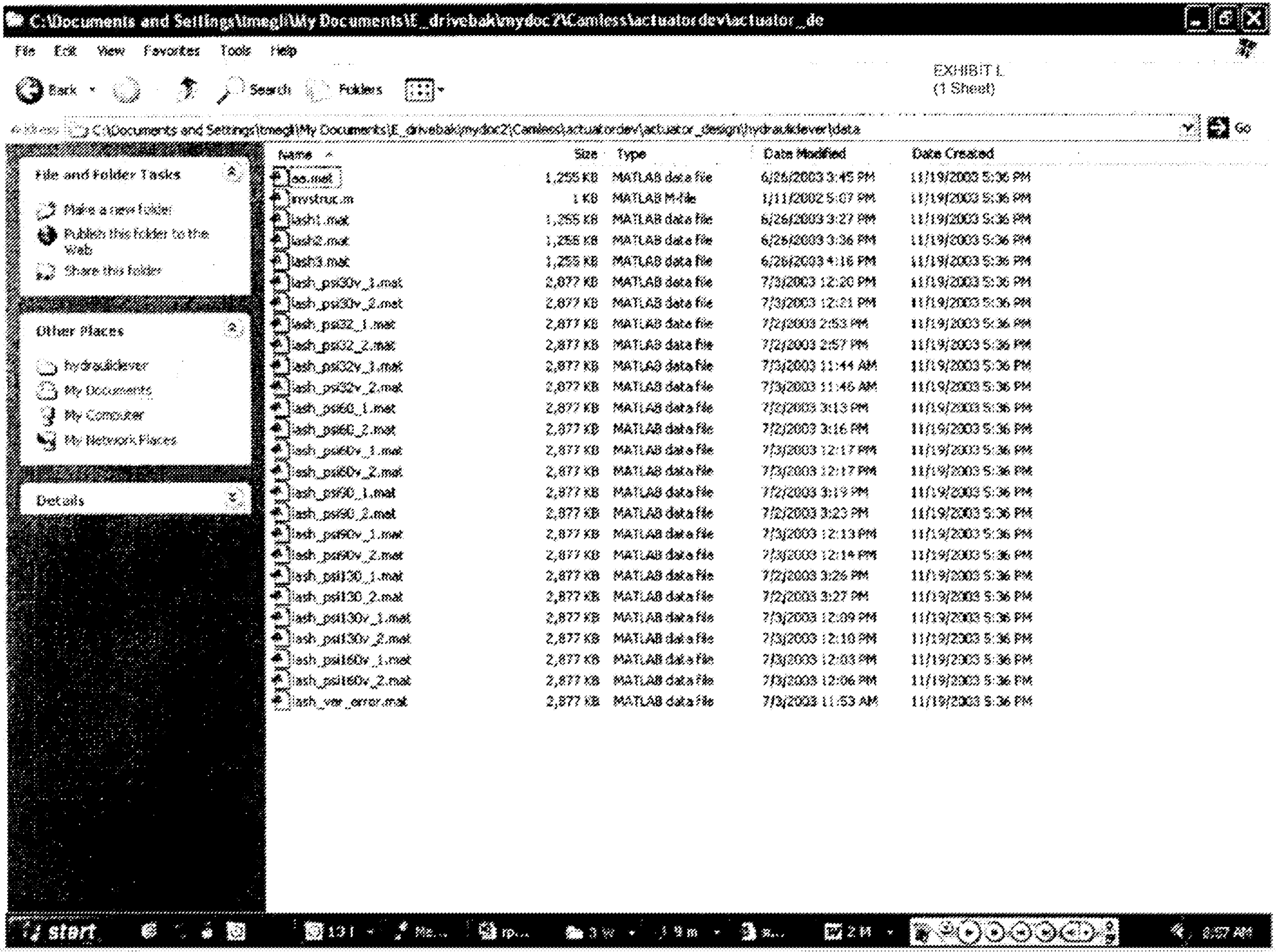
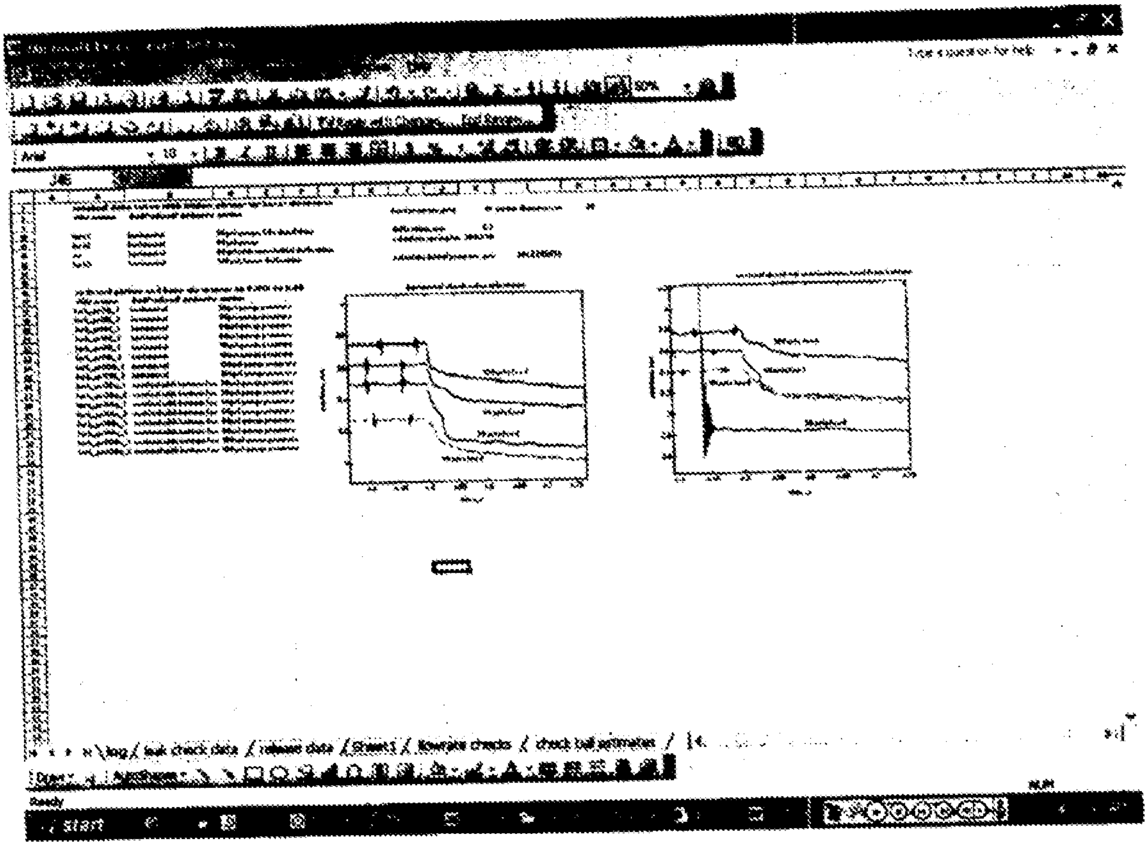
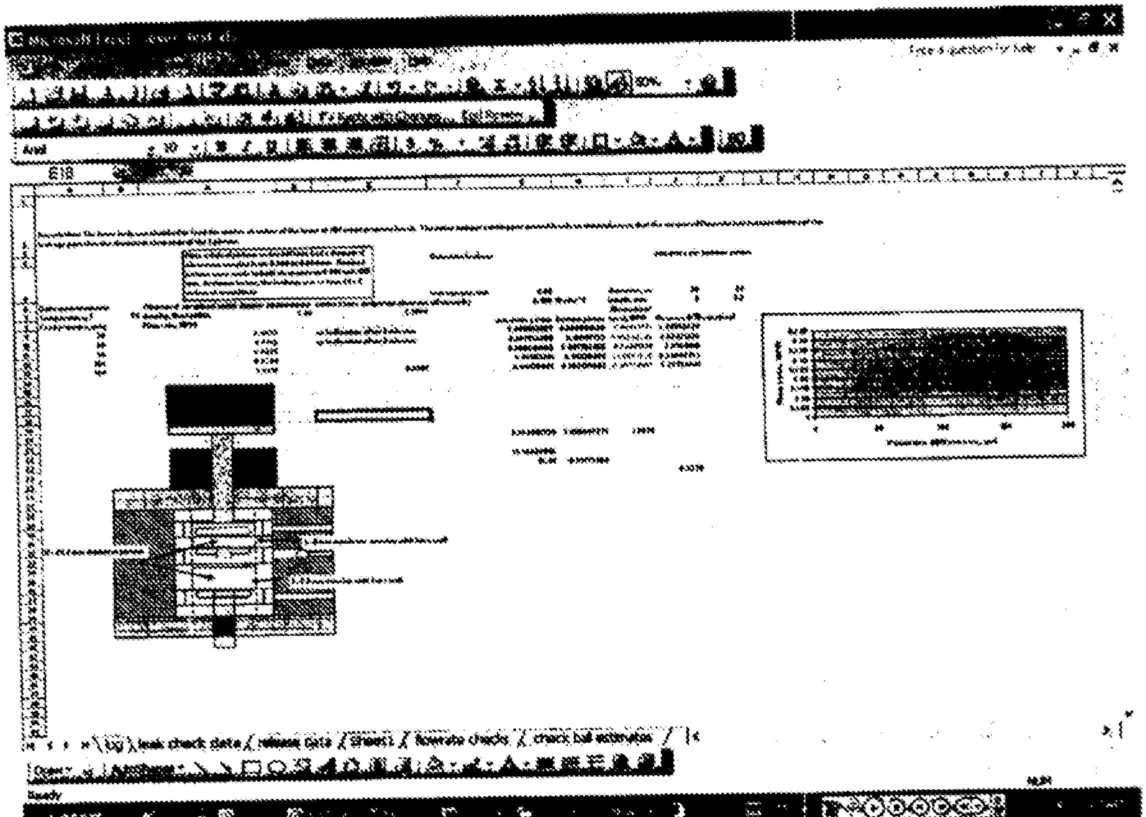


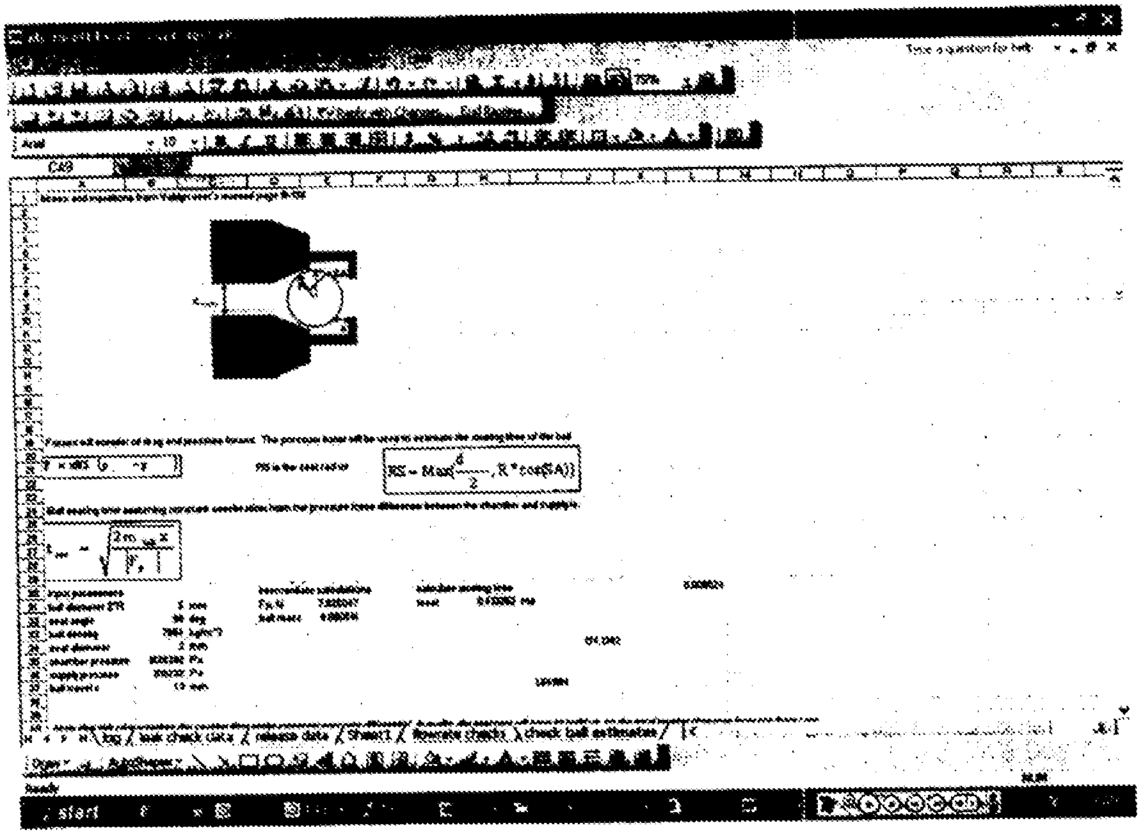
EXHIBIT M
(4 Sheets)



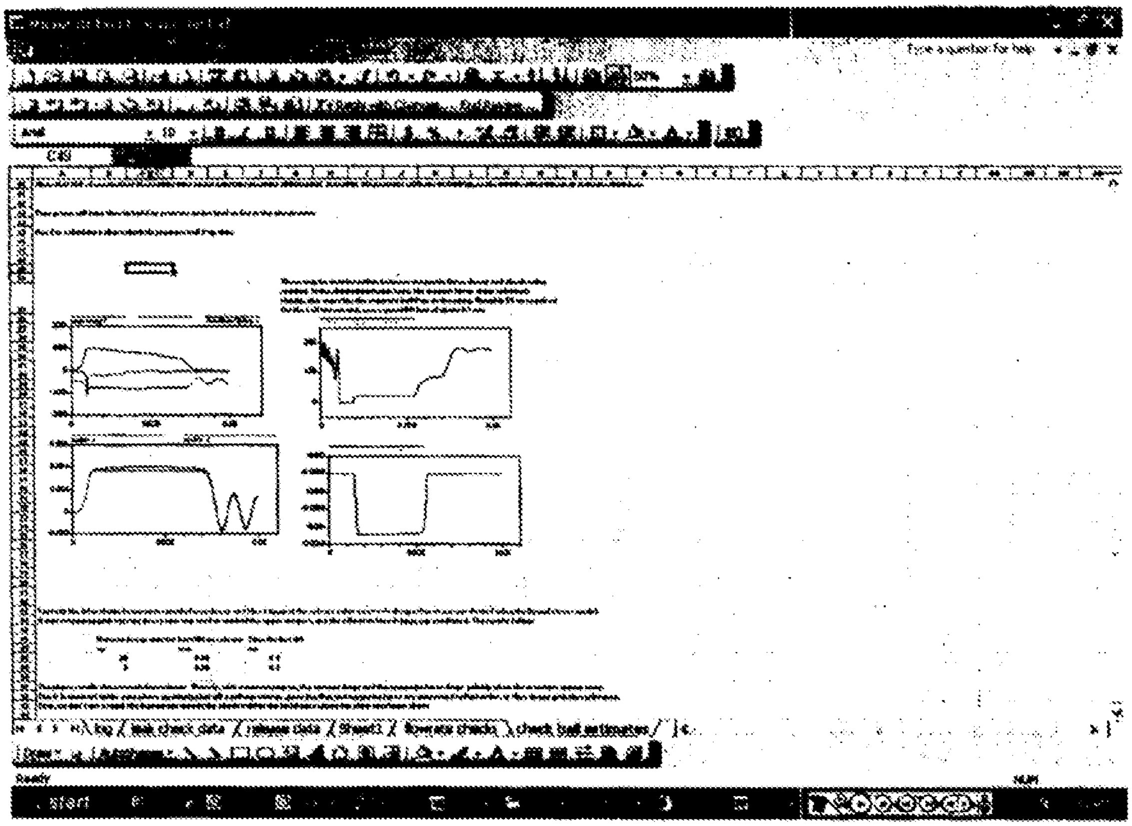
Another worksheet with test data



More data and analysis on leakage flow



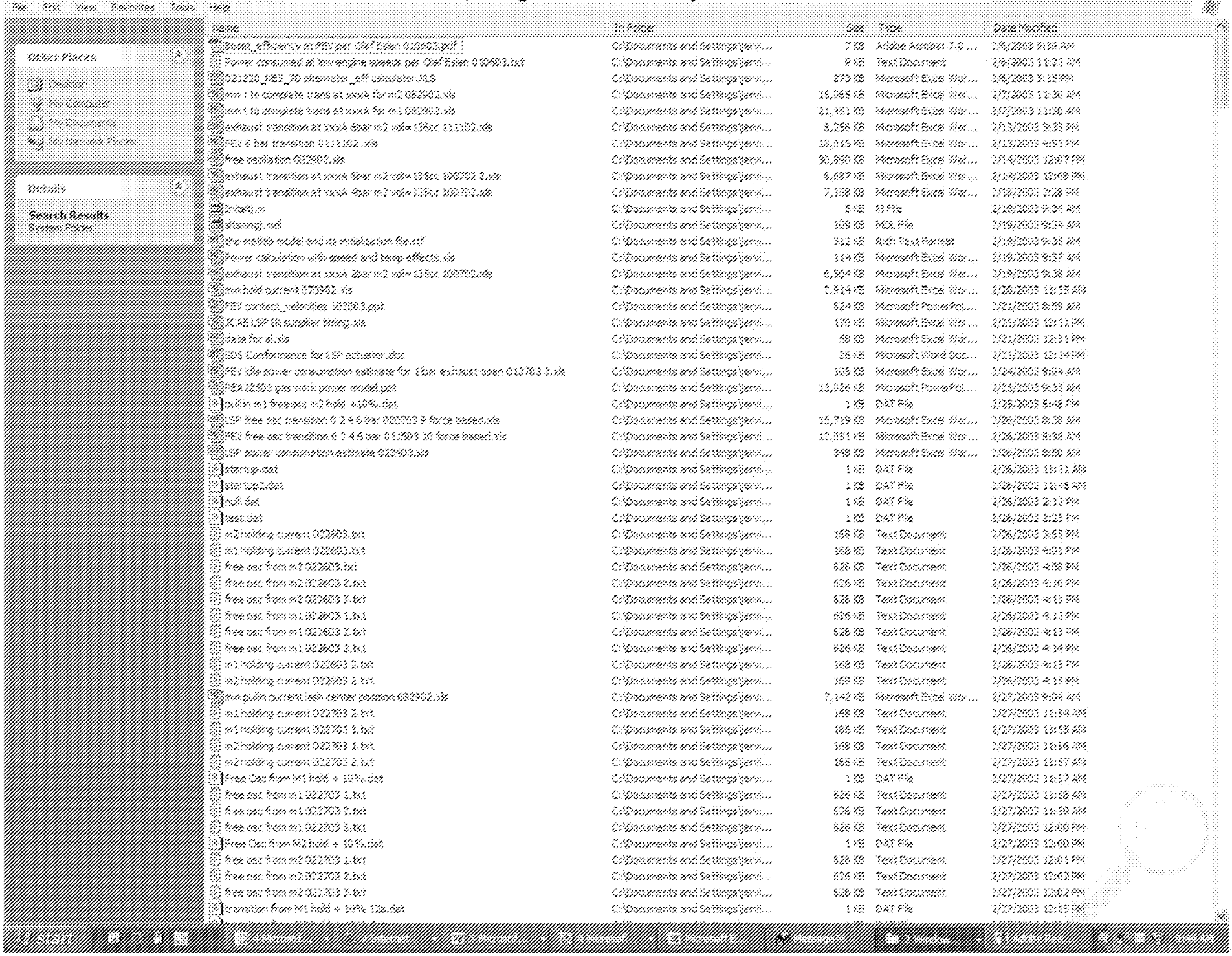
Check ball design analysis



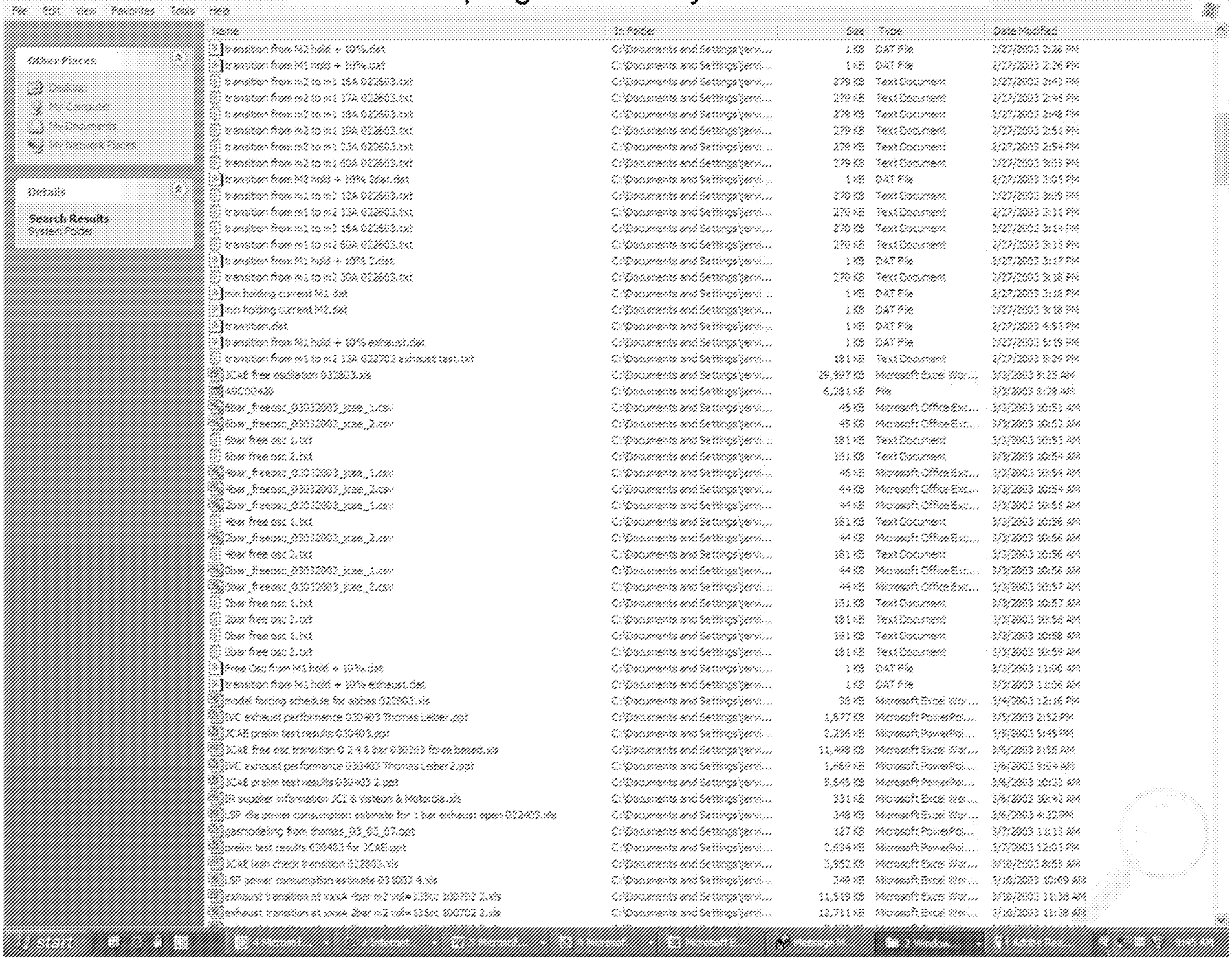
Dymola modeling results of prototype actuator with lever.

EXHIBIT N
(11 Sheets)

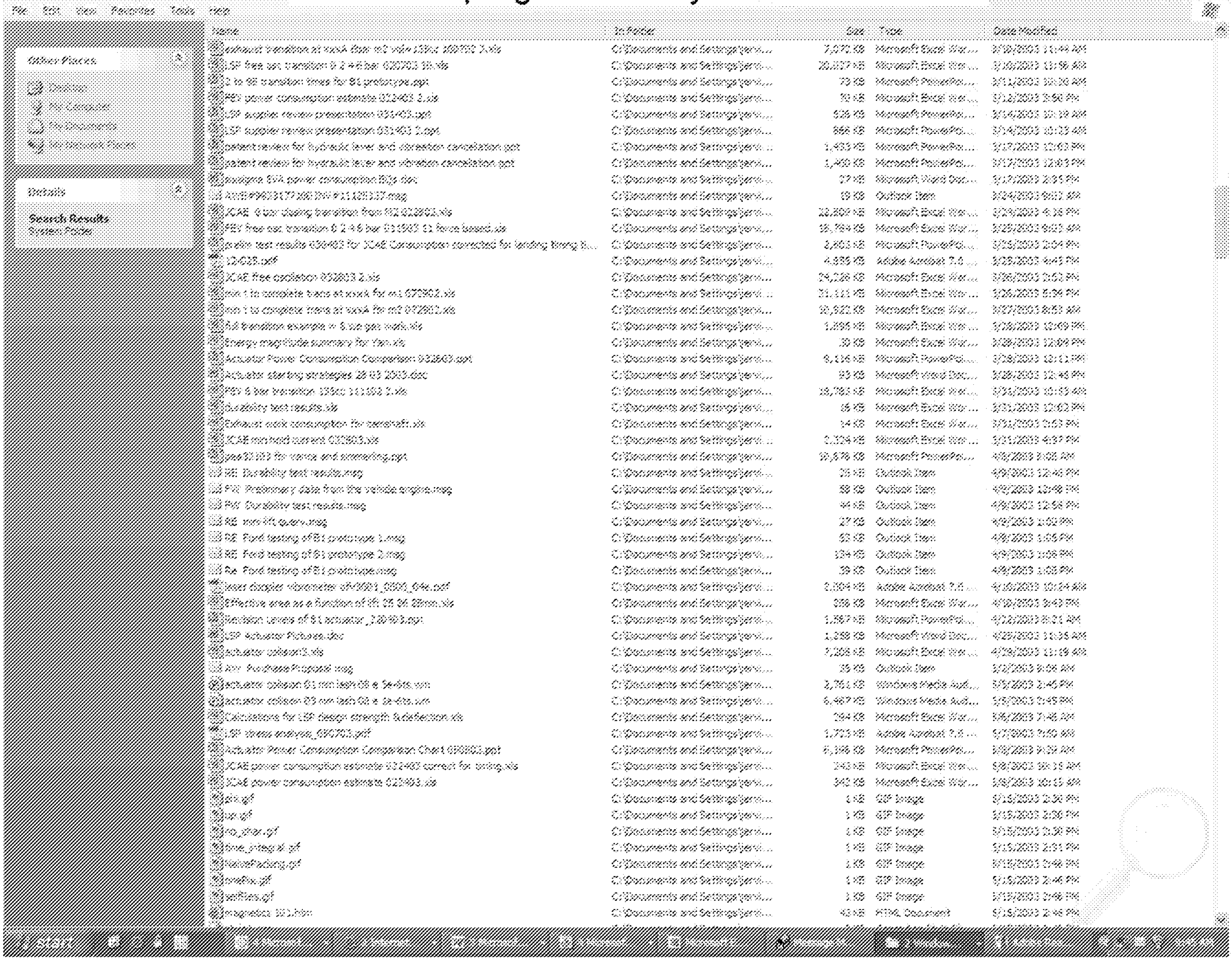
Ervin iEVA program activity from 2/6-2/27/03



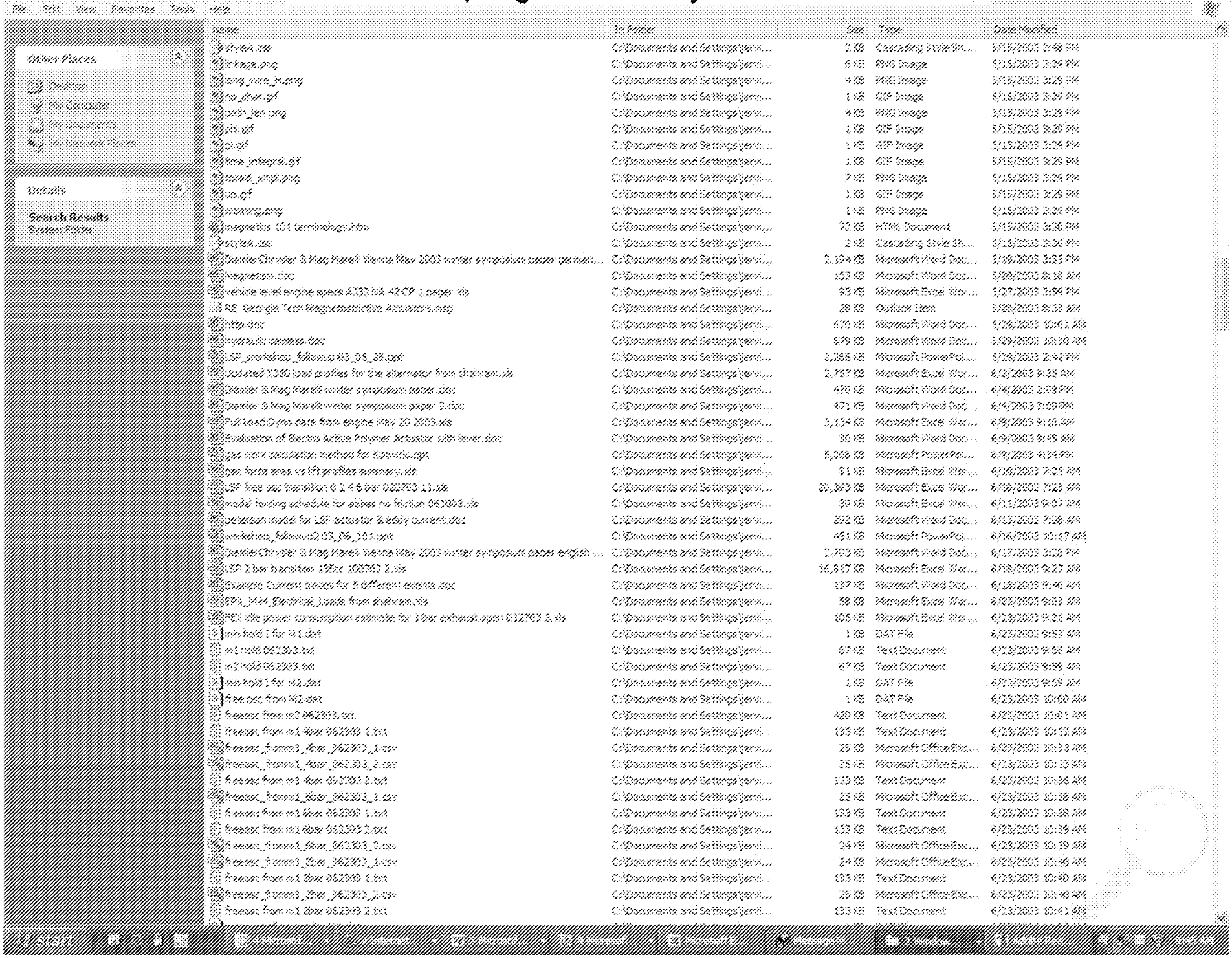
Ervin iEVA program activity from 2/27-3/10/03

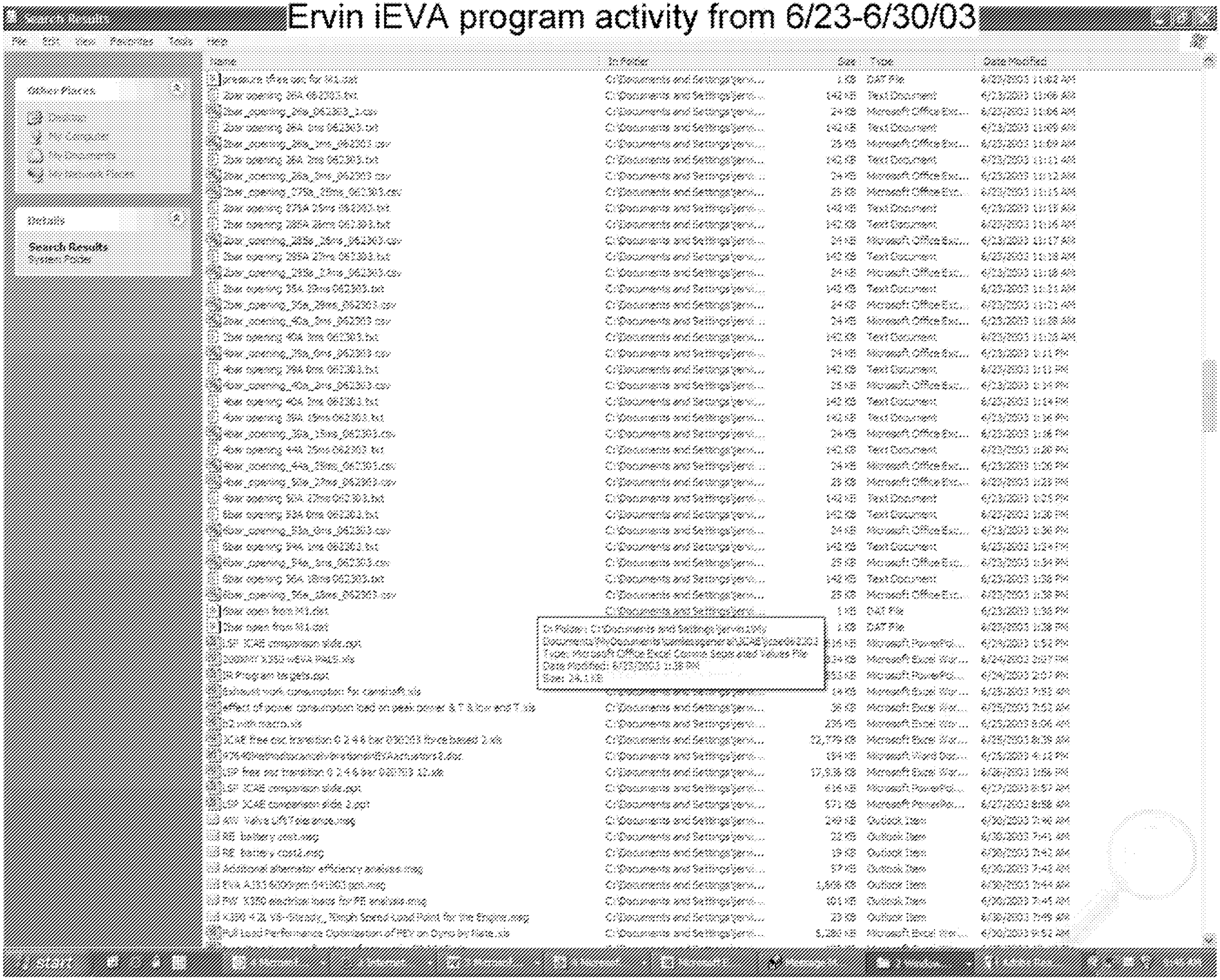


Ervin iEVA program activity from 3/10-5/15/03

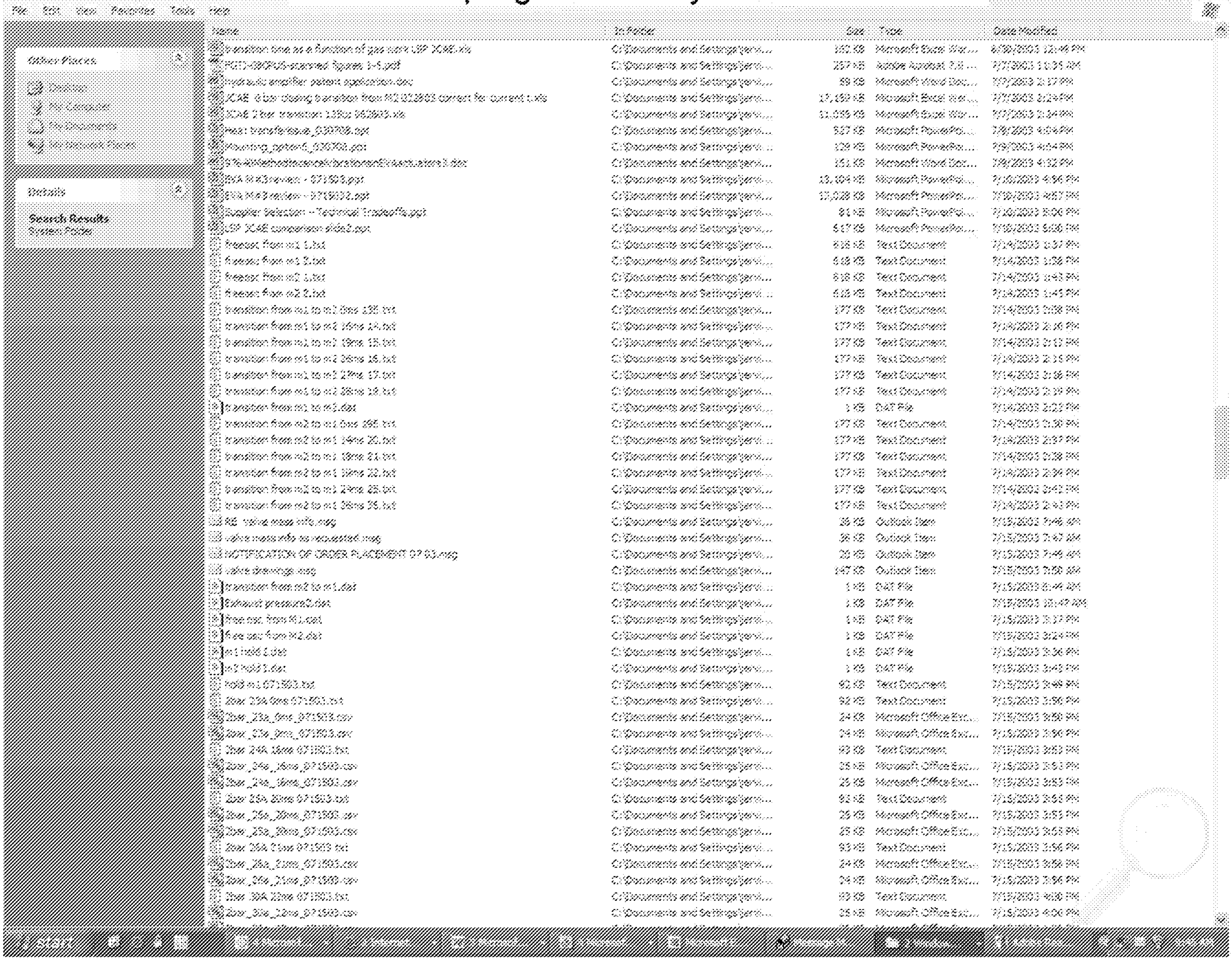


Ervin iEVA program activity from 5/15-6/23/03

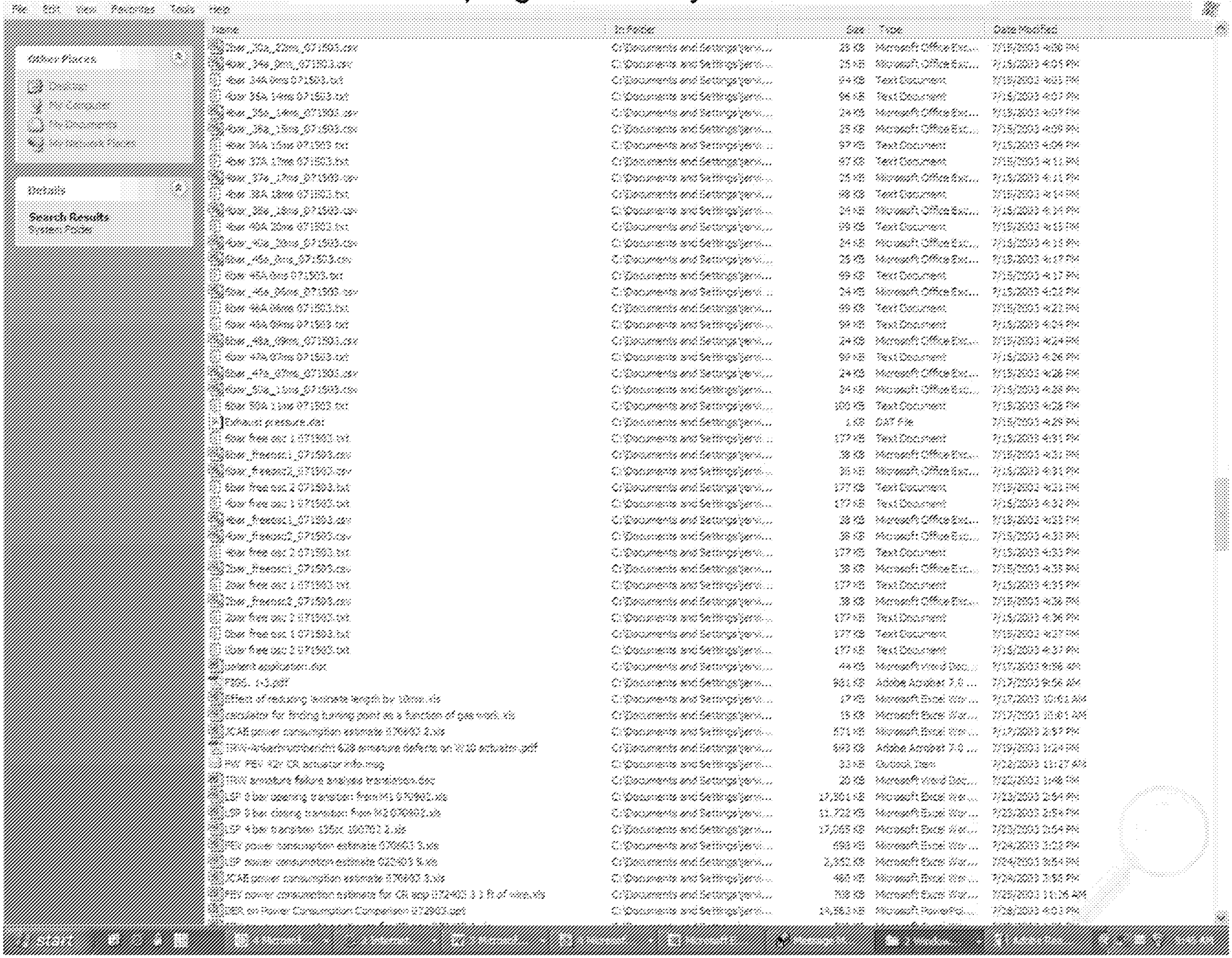



Ervin iEVA program activity from 6/23-6/30/03

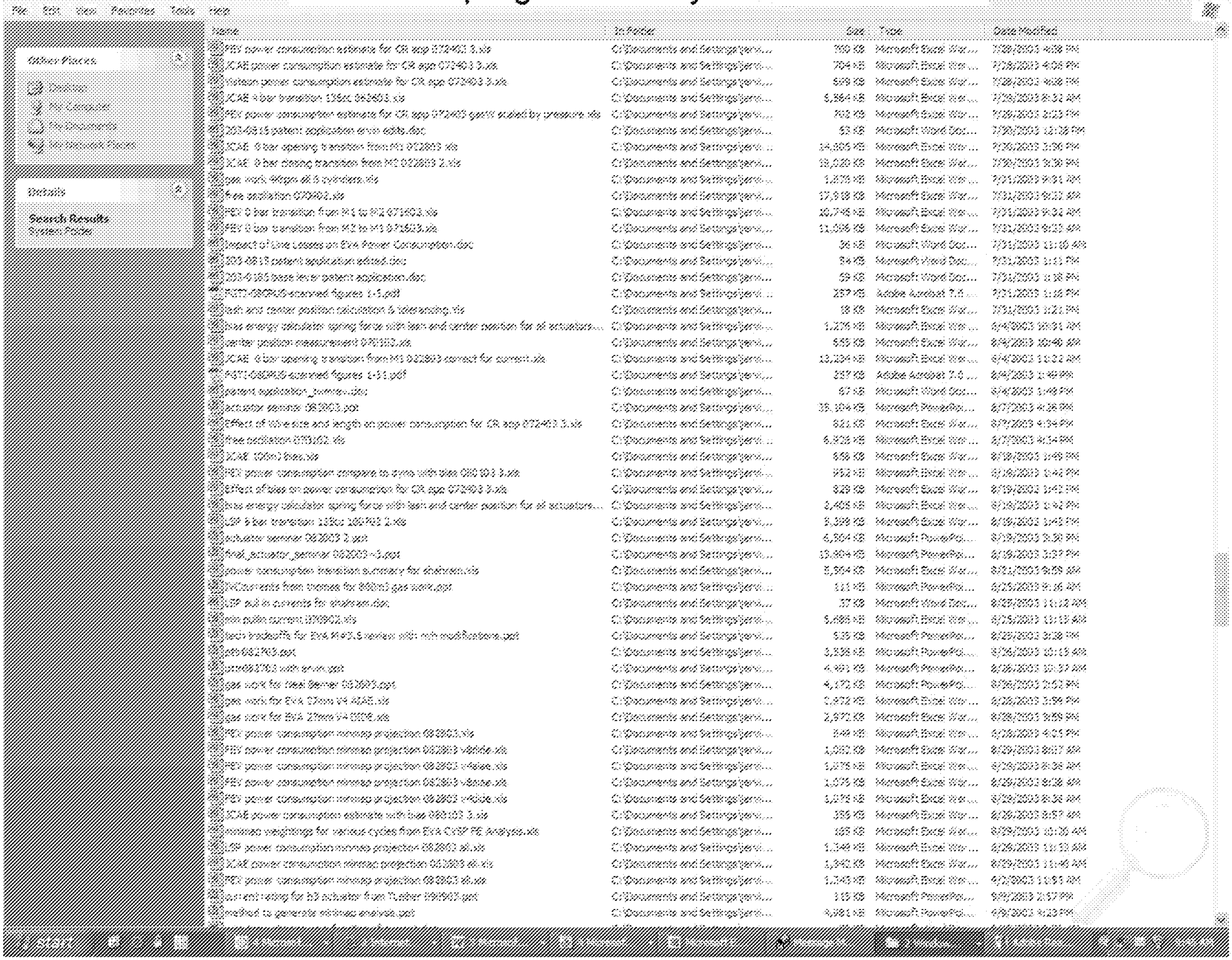
Ervin iEVA program activity from 6/30-7/15/03



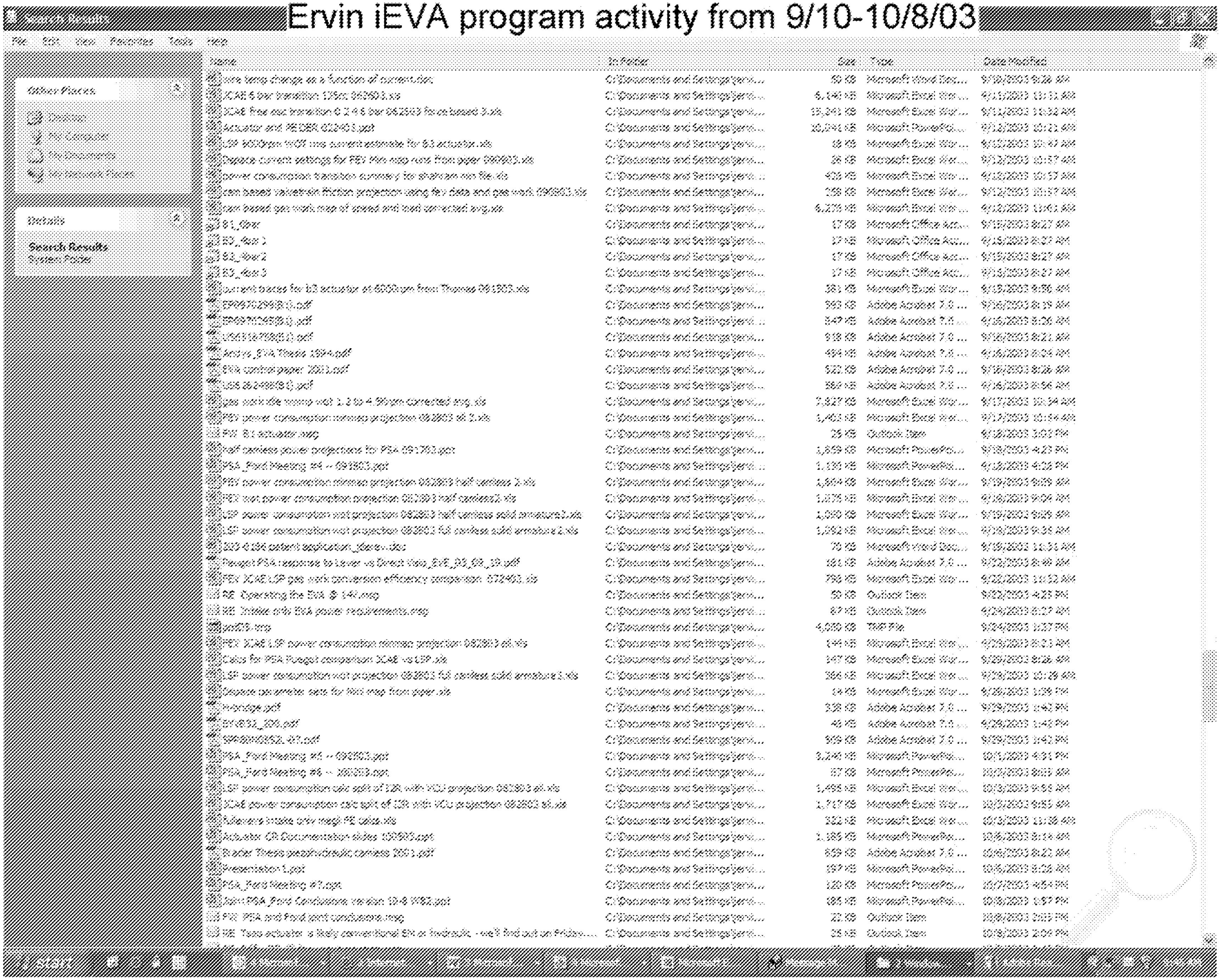
Ervin iEVA program activity from 7/15-7/28/03

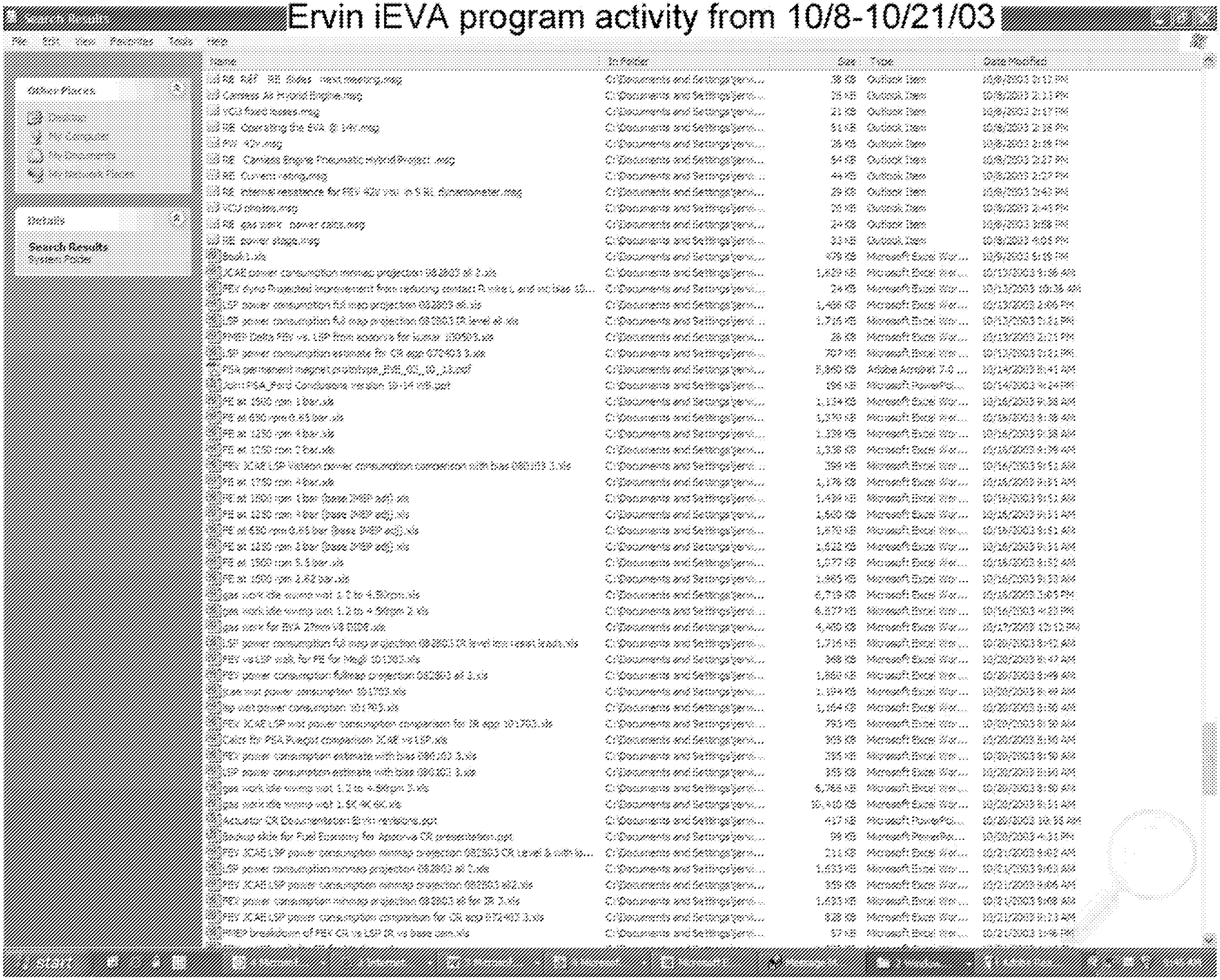


Ervin iEVA program activity from 7/28-9/9/03

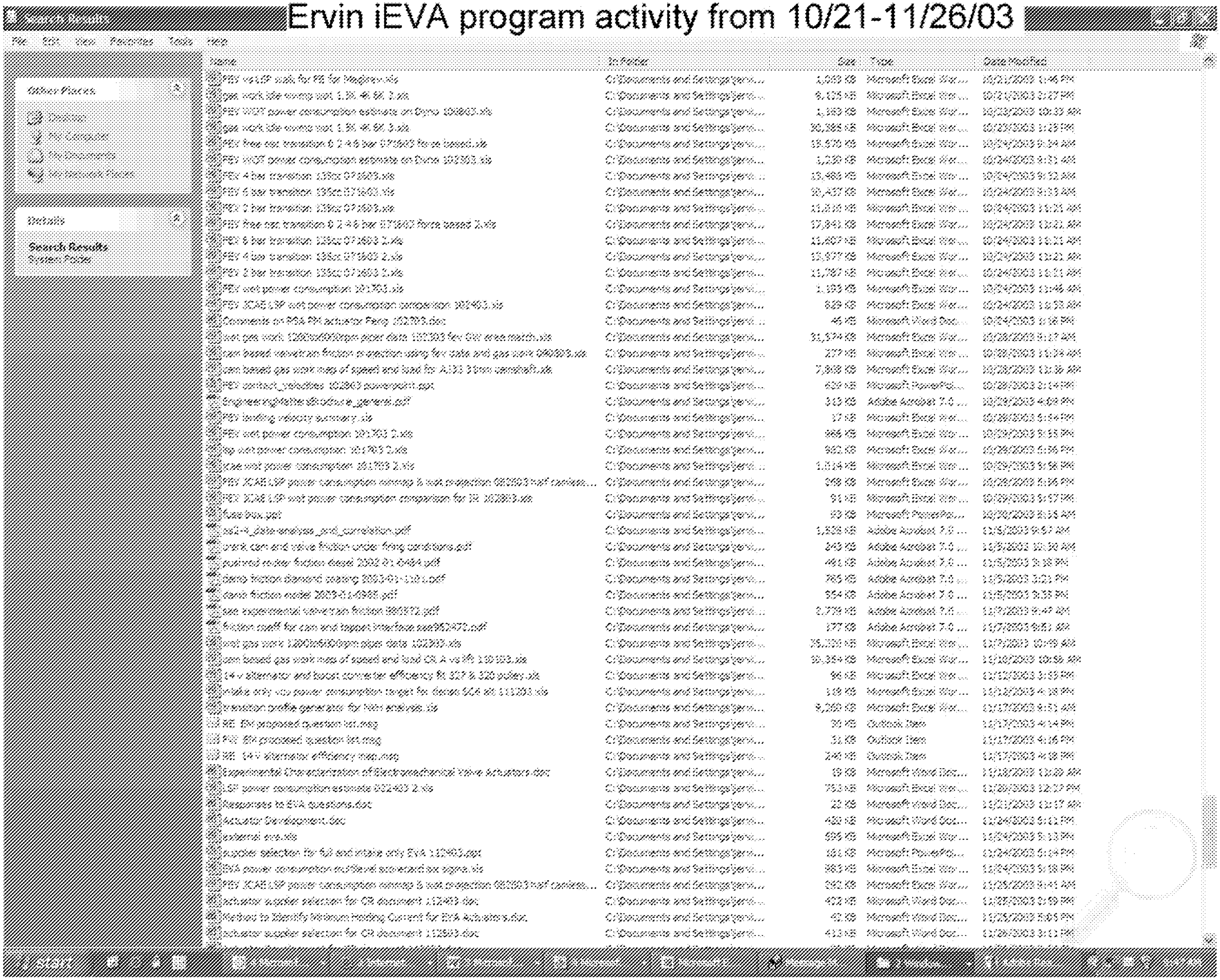


Ervin iEVA program activity from 9/10-10/8/03




Ervin iEVA program activity from 10/8-10/21/03

Ervin iEVA program activity from 10/21-11/26/03



Ervin iEVA program activity from 11/24-1/20/04